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STUDY OF PASSIVE TEMPERATURE AND HUMIDITY CONTROL SYSTEMS FOR ADVANCED SPACE SUITS

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FOR THE

AMES RESEARCH CENTER
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

FOREWORD

This report covers research activities conducted from July 1st, 1966 to September 1st, 1967 for the purpose of developing methods for passive control of temperature and humidity for advanced space suits.

This study was sponsored by the Ames Research Center of the National Aeronautics and Space Administration at Moffett Field, California, under Contract NAS2-3817. Messrs. E. G. Lyman and James Blackaby of the Environmental Control Research Branch were directing this effort for the NASA/Ames Research Center.

The program was performed by the Environmental Control and Life Support (ECLS) Department of the Systems Engineering and Integration Division of TRW Systems. Mr. A. P. Shlosinger, Head of the ECLS Research Section, was the Principal Investigator and Program Manager. Valuable contributions in the performance of this program were made by Messrs. Edgar F. Jacobi, L. G. Neal, D. J. Wanous, Members of the Technical Staff; Godfrey Hwa, Engineer and Research Specialist; G. B. Patchell and R. G. Simko, Laboratory Technicians.

In the materials area, this program was supported by the Nonmetallic Materials Section of the Materials and Processes Department. Messr. H. M. Gonzales, Member of the Technical Staff, and John C. Kimerly, Senior Materials and Processes Technician of the Nonmetallic Materials Section made major contributions to the successful performance of this program.

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS (Cont'd)

LIST OF ILLUSTRATIONS (Cont'd)

STUDY OF PASSIVE TEMPERATURE AND HUMIDITY CONTROL
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by

Arnold P. Shlosinger

SUMMARY

Investigations were performed to develop techniques for control of temperature in an extravehicular space suit by passive means. These techniques are intended to be integrated with techniques for passive control of humidity in space suits, investigated previously under contract NAS2-2102.

The techniques investigated are based on the use of the external suit surface as thermal radiator for rejection of excess metabolic heat and on a space suit shell of controllable overall thermal conductance. The controllable thermal conductance is achieved by a system of thermal insulation which is bypassed by devices similar to "Heat Pipes", modified to provide controllable heat flow rates and geometries applicable to a space suit.

Theoretical and experimental investigations demonstrated feasibility of the concept of a variable thermal conductance heat pipe. Concepts for the integration of a variable thermal conductance heat pipe into the fiberglass shell of a hard space suit were developed and fabrication techniques generated. Details of fabrication processes are discussed in a separate report, "Materials Research Report-First Year", TRW Systems No. 06462-6003-R000, September, 1967 prepared under the same contract, and are only briefly summarized in this report.

INTRODUCTION

Background

The subject of the study program covered by this report is the demonstration of feasibility and development of techniques for passive control of temperature in extra vehicular space suits. These activities were intended to continue research previously performed under contract NAS2-2102, which demonstrated feasibility of passive control of humidity in space suits (Ref. 1). Both techniques will eventually be intergrated into a Passive Temperature and Humidity Control System for Space Suits.

Present concepts of extra vehicular space suits rely for body temperature and humidity control on forced closed loop circulation of a liquid and/or a ventilating gas. Heat rejection is accomplished by control of temperature and humidity of the circulating fluids by means of a water evaporator or sublimation plates, located in a back pack. The approximate 2.5 to 3 m^2 (25 to 30 square feet) of external suit surface area are not used for temperature control by radiation to space. In fact, the man's body must be thermally well insulated from the outer suit surface. This is necessary because of extreme temperature gradients which exist between the solar irradiated and non-irradiated areas of the outer suit surface and because of variation in radiative heat inputs into the external suit surface, resulting from changes in the attitude of the astronaut relative to the sun or other radiant heat sources, or from shadowing of the astronaut by planetary bodies.

The thought of using the external suit surface for radiative rejection of heat has existed for considerable time. K. R. Cramer and T. F. Irvine, Jr. (Ref. 2, 3, and 4) performed thermo-analytical investigations of an astronaut in earth orbit. The study demonstrated the need for lateral temperature equalization on the external space suit surface and for controlled variations of the heat flow from the man's skin to the space suit external surface.

The use of the external suit surface as a thermal radiator is highly attractive. Man is a generator of significant amounts of body heat, which requires disposal. Evaporative cooling, according to present concepts, incurs a significant weight penalty for water, which is boiled off or sublimated to space vacuum. Power is required for forced circulation of heat transport fluids. If radiant heat emission from the external suit surface can be used to reject a major fraction of the metabolic heat and a means is found to transport heat from the man's skin to the outer suit surface without power penalty, savings in weight for water and batteries will be accomplished. For extended time extra vehicular activities (E. V. A.) such as lunar surface research or orbital assembly, these savings can be expected to be significant.

Application of the external suit surface as thermal radiator will require solutions to two problems:

- A means must be found to provide a high rate of heat diffusion laterally over the external surface of the space suit in order to reduce temperature gradients which would otherwise exist between solar irradiated and non-irradiated space suit surfaces.

- A means must be found to vary the effective thermal conductance between the astronauts skin and the outer suit surface in order to permit adaption to variations in metabolic heat production and changes in the external radiation environment.

A concept for passive control of the temperature inside a space suit was generated. In accordance with instructions received from the NASA-Ames Research Center, this concept was directed towards application to a hardshell, constant-volume-joint, space suit. The basic construction of the space suit shell was stipulated to be a reinforced fiberglass structure as shown in Figure 1. The internal reinforcement was, in the interest of better heat transfer, later modified to the configuration shown in Figure 2.

The task to be performed was to modify this space suit shell into a variable thermal conductance device which would, as required, provide either high thermal conductance or effective thermal insulation between the skin of the man and the outer suit shell surface.

The basic fiberglass space suit shell in accordance with Figure 1 and 2 will not be able to provide sufficient thermal insulation under a typical "cold" condition. Supplemental multiple reflective shield type vacuum insulation will be required. It was decided to make the basic fiberglass pressure shell as thermally conductive as possible. A thermal insulating layer would then be arranged external to the basic pressure shell of the suit. The thermal insulation would be by-passed by controllable heat transfer devices to provide an overall thermal conductance as required by astronaut metabolism and external radiation environment.

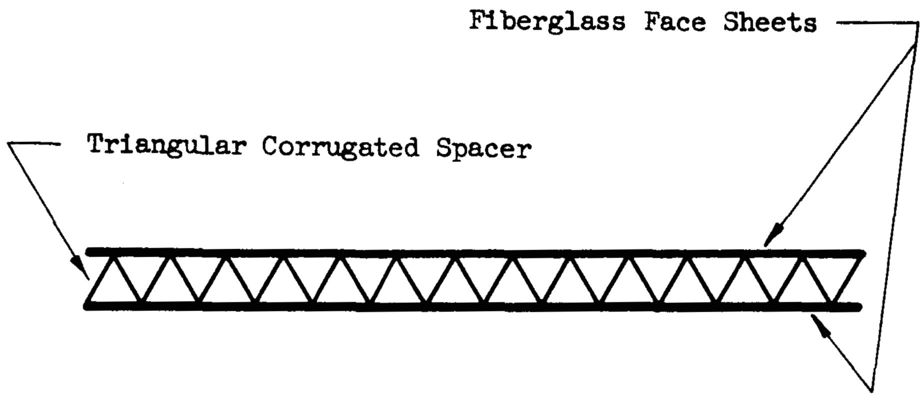


FIG. 1 SCHEMATIC CROSS SECTION OF FIBERGLASS SPACE SUIT SHELL AS ORIGINALLY PROPOSED

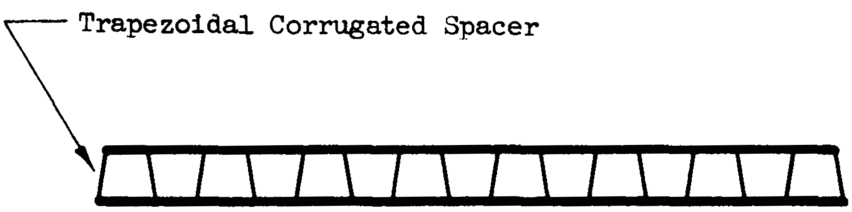


FIG. 2 SCHEMATIC CROSS SECTION OF FIBERGLASS SPACE SUIT SHELL AS MODIFIED FOR IMPROVED OVERALL THERMAL CONDUCTANCE AND APPLICATION AS "HEAT PIPE"

A technique recently developed and widely publicized for passive heat transport was taken under consideration to provide the basis for development of a controllable heat transport system. This technique consists of heat transport by mass transfer in a closed evaporating-condensing cycle using a capillary structure (wick) for return of the condensate. Experimental devices built commonly consist of a tube or pipe, lined on the inside with a wick, evacuated of noncondensible gas and supplied with just enough of a working fluid to completely soak the wick. Transport of heat in such a device occurs as a result of endothermic evaporation of the liquid phase of the working fluid at the warm end of the pipe, diffusion and flow of the vapor phase of the working fluid to the cool end, where condensation with release of heat takes place. Heat is thereby transferred from one end of the device to the other.

Because of the high heats of evaporation of potential working fluids, which include water, liquid metals, ammonia, alcohols and others, very high heat transport rates have been observed. The devices built and studied, using these principles, are commonly referred to as "Heat Pipes."

The thought occurred that if a heat pipe type device could be made controllable in response to an applied signal, it would provide the potential for the development of a variable thermal conductance device suitable for control of heat transfer across the space suit shell. The problem of developing a passive temperature controlled space suit then reduced itself to the performance of a number of specific tasks which include:

- Development of a controllable heat pipe type device
- Development of controllable heat pipe type devices having geometries compatible with the geometry of a space suit.
- Selection of suitable working fluids, capillary materials and construction materials for the variable heat transport devices.
- Development and generation of fabrication techniques which will permit the fabrication of a space suit shell with integrated controllable heat transport devices and adequate thermal insulation.

The Controllable Heat Pipe Concept

The first publication on a heat pipe was probably an article which appeared in the Journal for Applied Physics in June, 1964, entitled, "Structures of Very High Thermal Conductance" by G. M. Grover, T. P. Cotter, and G. F. Erickson, from the Los Alamos Scientific Laboratory. This article described the construction and the results of several experiments with a heat pipe.

It was reported in this article that nearly isothermal conditions where observed at high heat flow rates, leading to the conclusion that there was no measurable temperature gradient inside a heat pipe between the evaporating and condensing surfaces of the liquid-soaked wick. This can be explained by the fact that, while vapor transport in a heat pipe results from a pressure difference between the warm and the cold end of the device, this pressure difference, because of the large vapor flow area, is very small. Therefore, the evaporating and condensing temperatures, which are functions of the pressures existing at the evaporating and condensing end, will be very close to each other.

The experimentation was performed with, what could be called, a "Classical" heat pipe, consisting of an actual pipe or tube, enclosing the capillary structure and the working fluid. The process which leads to heat transport in a heat pipe is, of course, not dependent or related to a specific geometry. Any cavity, free of noncondensible gases and lined with a wick which is saturated with a liquid of a vapor pressure suitable for the operating temperature range of the device, will transport heat by the mechanism of a heat pipe. The term "heat pipe", in this report, is used in the broad sense of an evaporating-condensing cycle with condensate return by capillary action, even if the geometry of the devices discussed has no similarity with the geometry of a "pipe".

In order to make a device of this type controllable it is necessary to control one of the mechanisms on which heat transport in a heat pipe depends. Several possibilities were considered. It had been observed that a noncondensible gas, present in a heat pipe, is flushed by the working fluid vapor to the cool end of the heat pipe, where it collects and effectively interferes with heat transfer. This phenomenon has the potential of application to heat flow control in a heat pipe. Whenever reduction of heat flow is desired, a controlled amount of noncondensible gas could be introduced into the heat pipe. Whenever increased heat transfer rates are required, the noncondensible gas could be bled off from the cool end, restoring full heat transport capability.

There are several reasons why this technique has not been considered as practical to applications in a space suit. Venting the noncondensible gas to space vacuum would incur the need for storage and replacement of an expendable gas. Each time venting takes place, a small amount of the working fluid vapor, which has diffused into the noncondensible gas layer, would also be lost. This would be directly contrary to the intent of this program to develop a technique requiring less expendable materials for space suit operation.

A technique generated in preliminary studies of the applicability of heat pipe type devices to heat transport in a space suit would control the evaporating temperature by control of the pressure inside the heat pipe. This requires that the volume of the heat pipe cavity be made variable, such that constant pressure, within required operational limits, can be maintained within the heat pipe cavity under variations of heat flow into the device. In Figure 3 a bellows is shown attached to the heat pipe. If this bellows is externally loaded with a constant gas pressure, it will, when heat is applied to the warm end of the heat pipe and vapor is generated as a result of this heat addition, expand and maintain the pressure at a level dictated by the externally applied control pressure. This would maintain the evaporating temperature at the saturation temperature of the working fluid corresponding to this pressure. As long as the source of heat, i.e., the skin of the astronaut, is at a temperature equal or lower than this temperature, heat transfer from the skin would cease. Thereby, the skin temperature of the man could be maintained at a level dictated by the control pressure applied to the bellows and heat removal from the body adjusted to physiological requirements. Figure 4 shows an arrangement which would apply this technique to flexible heat pipes for a soft suit. It would, however, require an undesirably complex system of dual layer heat pipes, distributed over most of the area of the space suit surface.

Two other techniques would control heat transport in a heat pipe. If the wick would be arranged such that the continuous capillary flow path provided can be interrupted (Figure 5), the mass transfer loop in the heat pipe would be broken and transport of heat would cease. The other technique would stop the flow of vapor from the warm to the cool end of the heat pipe by the insertion of a valve in the vapor flow passage. This technique of control of the vapor flow passage has been selected as the most practical means of control of heat flow for space suit applications.

In order to stop the flow of vapor in a heat pipe, it will be necessary to separate the vapor flow passage from the wick passage as shown in Figure 6. The wick passage will consist of a tube in which the wick is tightly inserted to prevent vapor bypassing. When all

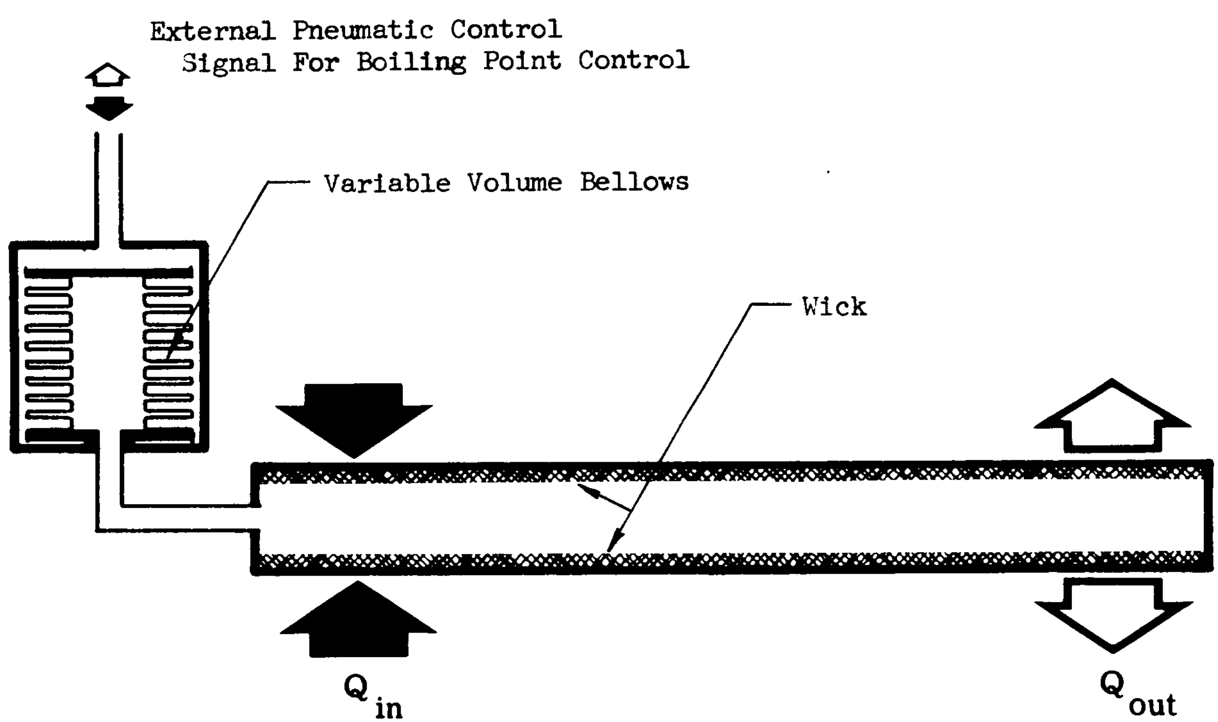


FIG. 3 CONTROLLABLE HEAT PIPE (VARIABLE VOLUME - CONTROLLED PRESSURE CONCEPT)

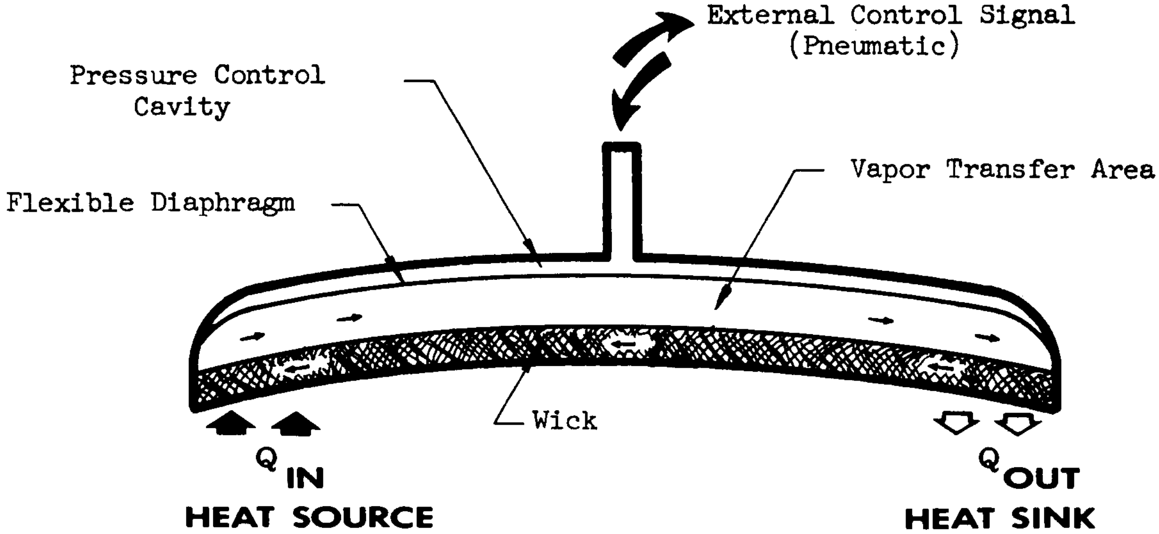


FIG.4 - CONTROLLABLE SOFT SPACE SUIT HEAT PIPE (VARIABLE VOLUME - CONTROLLED PRESSURE CONCEPT)

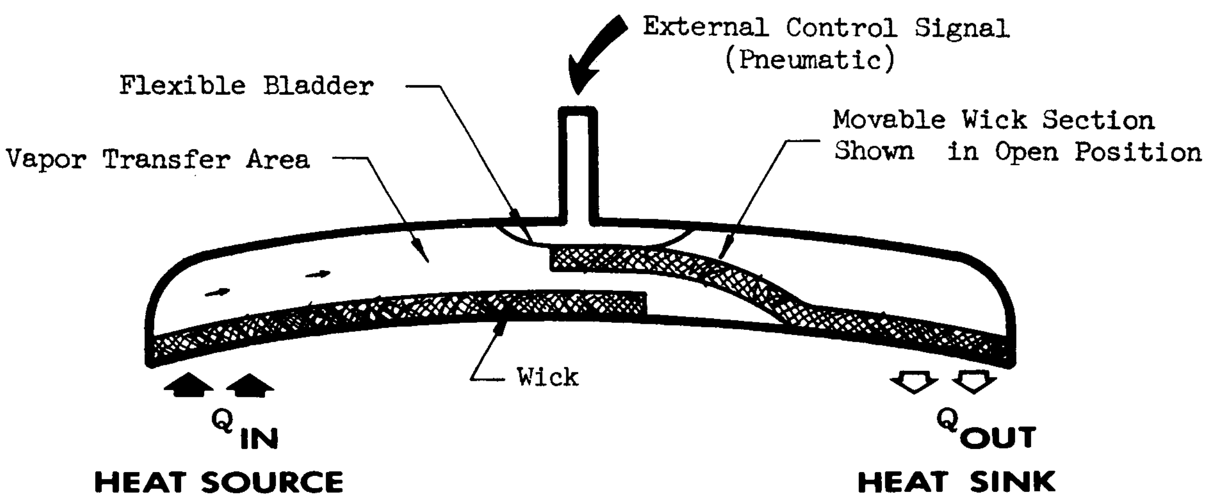


FIG.5 - CONTROLLABLE HEAT PIPE (CONDENSATE FLOW CUT OFF CONCEPT)

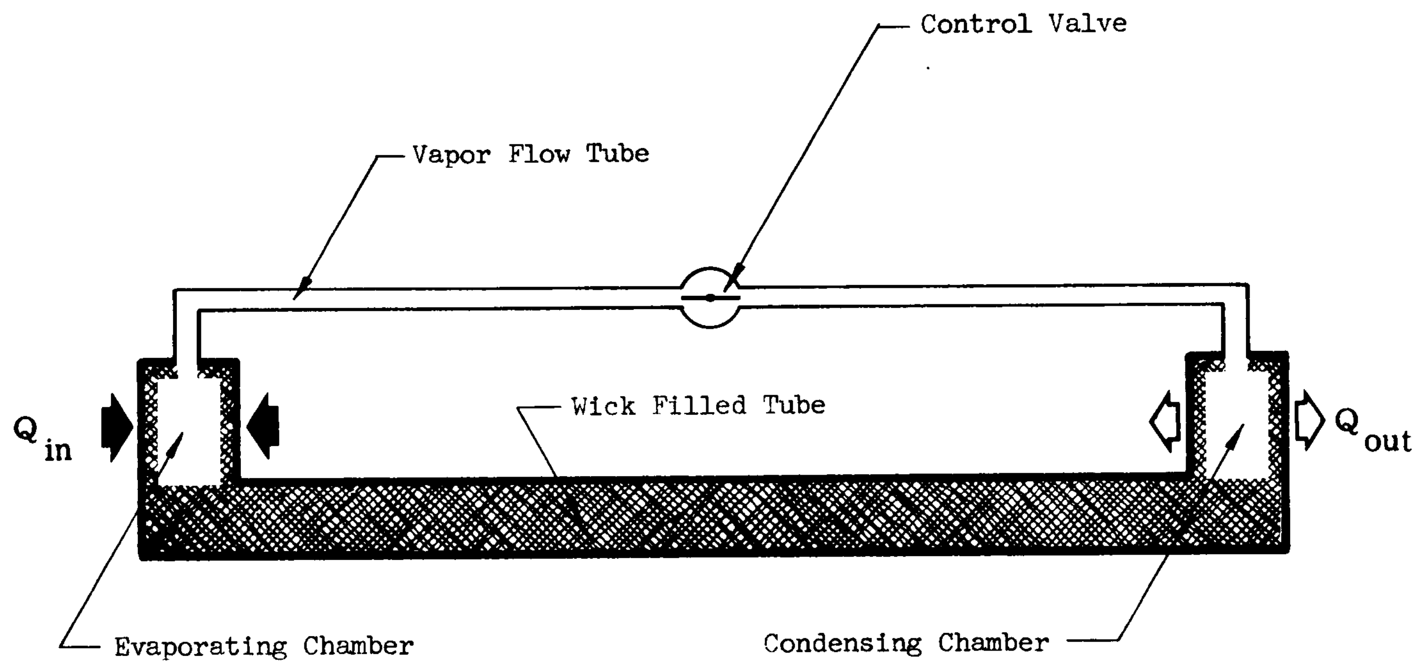


FIG.6 VARIABLE HEAT PIPE (VAPOR FLOW CONTROL CONCEPT)

pores in a wick are saturated with liquid, the wick is vapor non-permeable. The only vapor flow passage would then be through the control valve.

Advantages of this technique include the possibility to modulate the vapor flow by using a valve with throttling characteristics. As an effect of such throttling, an increasing pressure differential will be generated between the evaporating and the condensing chambers of the device. As evaporating and condensing temperatures are functions of the respective pressures, temperature regulatory functions can be achieved.

Further advantages of this technique are that only relatively small passages must be controlled and the control valves can be located at a convenient location. Excessive complexity is thereby avoided. Compared to interruption of the capillary flow in the wick, the advantages of this technique include modulating characteristics, faster response to a control signal and avoidance of the need for manipulating a wick every time a control signal occurs, which could result in breaking of fibers and deterioration of structural integrity of the wick.

The Variable Conductance Space Suit Shell Concept

Applying the previously described technique of controlling heat flow in a heat pipe by a valve in the vapor flow passage, a concept for a variable conductance space suit shell was generated. As previously stated, the construction of the basic structural and pressure shell of the hard space suit had been stipulated by the NASA-Ames Research Center. It was considered impractical to incorporate a variable thermal conductance heat pipe into the construction of this shell. This shell would, even at minimum thermal conductance, not be adequate to provide insulation under an eclipse condition or during the lunar night. It was, therefore, decided to use heat pipe technique to give this shell maximum effective thermal conductance. The purpose of this shell would then be to provide a pressure vessel and basic suit structure. On the outside surface of this shell, a layer would be provided in which all the devices providing thermal control would be included. These devices would include thermal insulation and variable conductance heat pipes to bypass this insulation, when transfer of heat from the body of the astronaut to the outer face of the external radiating surfaces of the suit is desirable.

Figure 7 shows a conceptual diagram of this shell as originally conceived. The inner pressure shell shown in this figure was modified, as earlier stated and shown in Figure 2. This modification

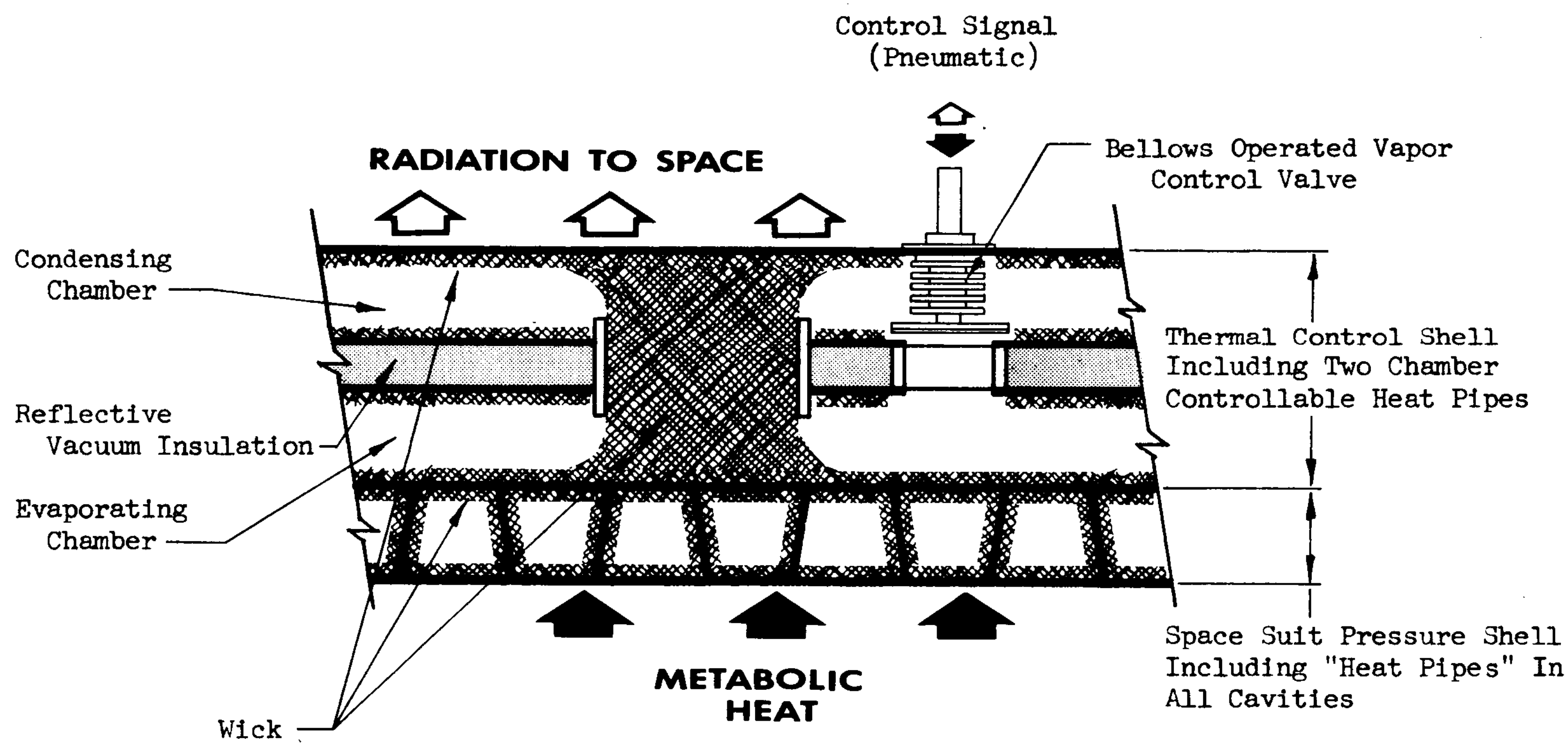


FIG. 7 SCHEMATIC CROSS SECTION OF A VARIABLE CONDUCTANCE SPACE SUIT SHELL

consisted of changing from the triangular reenforcing channels shown in Figure 1, to trapezoidal channels shown in Figure 2. The reason for this change was that, with the triangular arrangement, conductive heat transfer through fiberglass layers and wicks will occur in three places between the two boundary surfaces of the inner shell. With the trapezoidal arrangement, conductance heat transfer is required only in two places.

The outer face of the inner shell provides part of the shell of a two-chamber heat pipe which constitutes the outer space suit shell. The outer space suit shell consists of two cavities which are separated by multiple reflective shield-type vacuum insulation (super insulation). This insulation layer will provide protection against excessive heat loss when the heat pipe of the outer suit is in the "shut" position. Whenever heat flow from the body of the astronaut to the external surface of the space suit is desired for purposes of heat rejection, heat will be transferred from the skin of the astronaut, through the inner suit shell heat pipes and the outer suit shell controllable heat pipes to the external radiating surface of the suit.

The concept as described, requires in its application to a hard suit, that heat be transferred from the skin of the astronaut to the inner face of the pressure shell by means not requiring direct contact between the hard suit shell and the skin. A hard space suit can not be made tight fitting. Space must be provided between the human body surface and the inner suit surfaces for body expansion resulting from breathing, muscular activities, etc. Radiative heat transfer for this application can only provide a part of the heat transfer required. It will, therefore, be necessary to provide flexible skin contactors, which by means of either high thermal conductivity or heat pipe function, transfer the heat from the skin of the man into the temperature controlling space suit shell.

In performance of this phase of the contract, this specific problem was only considered from the conceptual viewpoint. In mutual agreement with the Contract Monitor, it was decided to put the major effort into the development of the Variable Conductance Space Suit Shell and demonstrate its feasibility. It is believed that heat pipe-type devices of flexible materials, which may be flexible plastics or metallic bellows as diagrammically shown in Figure 8, have the potential to solve this problem.

The major problems for the development of the Variable Thermal Conductance Space Suit Shell was to develop and demonstrate a technique for heat flow control in a heat pipe by control of the vapor flow passage, develop a technique for fabricating the inner pressure

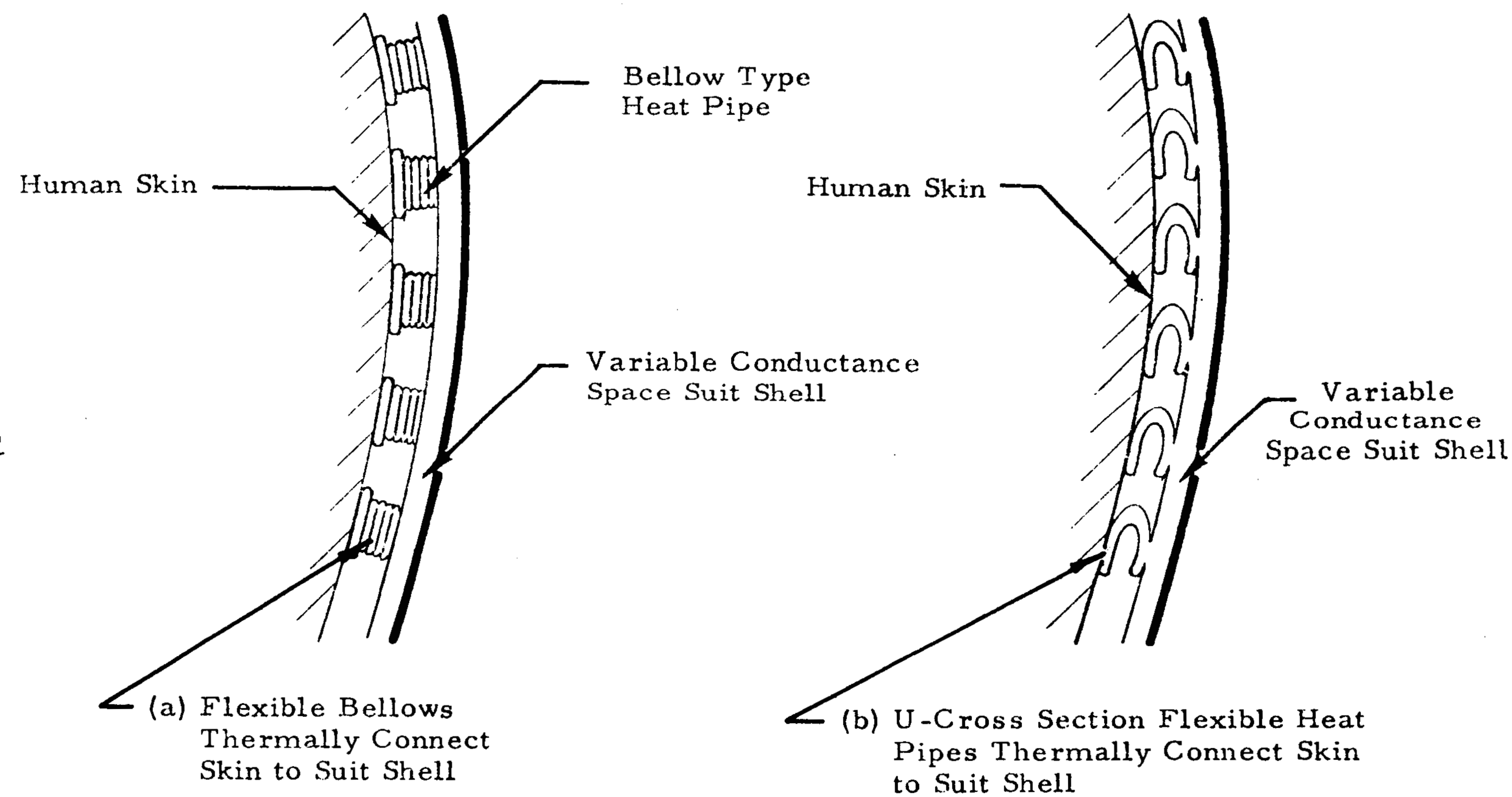


FIGURE 8 FLEXIBLE HEAT PIPE CONCEPT FOR HEAT TRANSPORT AND SUIT SHELL SPACING

shell such that the small channels, formed by the trapezoidal reenforcement, would be heat pipes and selection of suitable materials, including wicks, working fluids, bonding techniques and others for the fabrication of these devices. Analytical and experimental studies performed to solve these problems were the subject of the program and are documented in this and in a separate report entitled, "Study of Passive Temperature and Humidity Control Systems for Advanced Space Suits, Materials Research Report, First Year" which deals with material selection and fabricating processes.

ANALYSIS

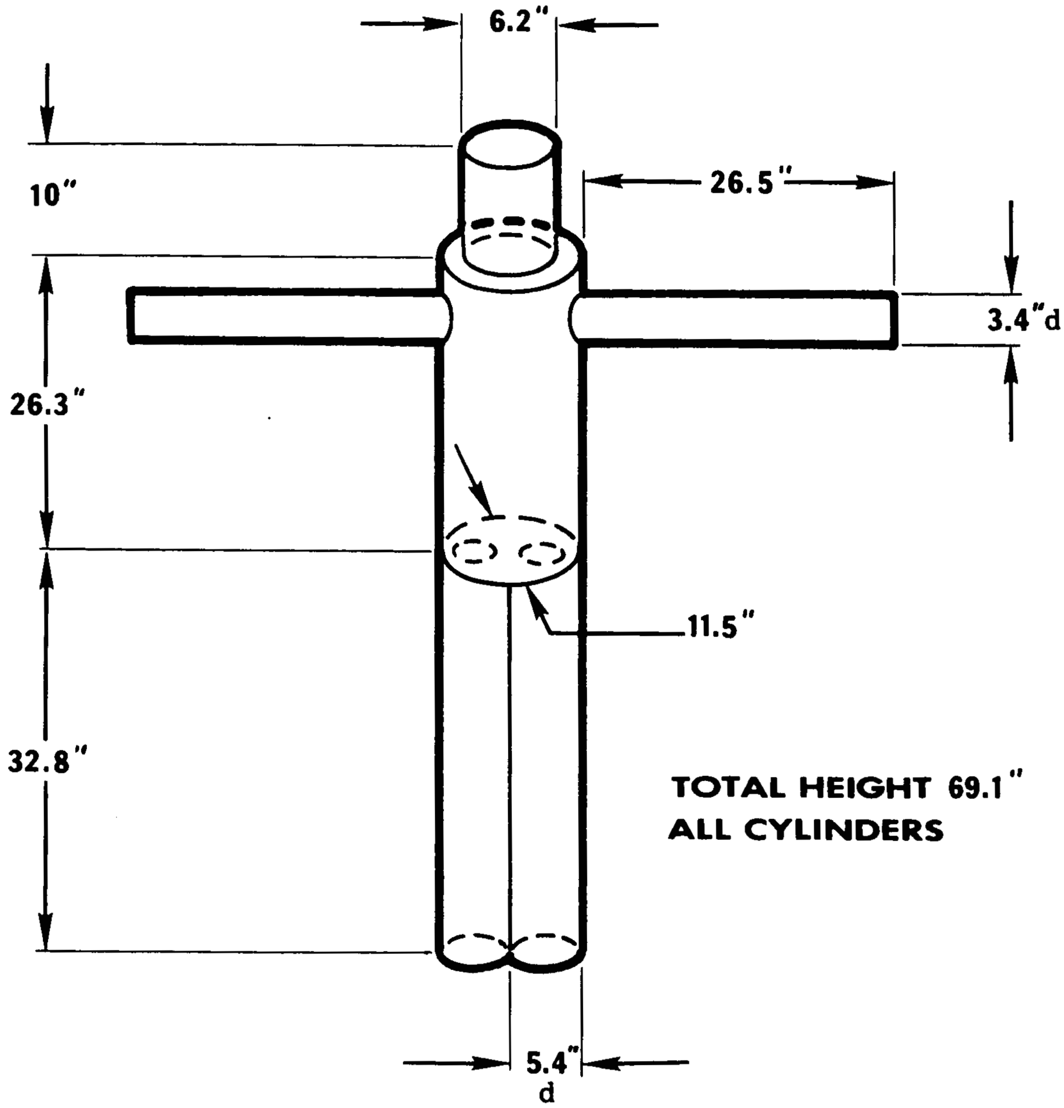
Space Suit Heat Rejection Capability

The purpose of this analysis was to determine how much heat can be rejected from the outer suit surface by radiation to space. The heat rejection capability of the outer suit surface depends on the temperature and the thermal emissivity of the external suit surface. The temperature will depend on the temperature gradient required to transmit heat from the skin of the man to the outer surface of the space suit. This temperature gradient has at this time not firmly been established by experiment.

The temperature of the outer suit surface will further depend on external radiant heat inputs, which will depend on the absorptivity of the outer suit surface to the wave length of the incident radiation, exposure, shadowing, and attitude relative to the radiation source which will determine the effective area receiving radiant heat inputs.

Several simplifying assumptions were made for purpose of this analysis in order to maintain a proper balance of effort between this analysis and the other research goals of this program.

A geometrical model of a suited astronaut (Figure 9) was used for determination of effective radiation areas of the suit. To allow for interreflections between limbs, and limbs and torso, it was assumed that one half of the leg area, one half of the arms, and an area on the torso equal to one half of the area of the arms would not be effective as thermal radiators. This assumption is conservative. Actually, radiation from the inside of the legs, from the torso under the arms and the side of the arms facing the torso will not be equal to zero. These surfaces will emit radiation to space. Only the view factor between these surfaces and space will be diminished by the presence of the adjacent body surfaces.



TOTAL SURFACE AREA = 21.1 SQ. FT.

FIG. 9 GEOMETRIC MODEL OF MAN
(FROM REFERENCE 2)

Data from Reference 5 (Table 2-8, "Gemini Suit Thermal Characteristics") were used as basis for an estimate of the external surface area of the space suit. This area is given as 2.83 m^2 (30.56 $ft.^2$). The area relationship between torso, legs, and arms was based on Figure 9 by proportioning the external area of the Gemini suit of 2.83 m^2 (30.56 $ft.^2$) in accordance with the nude body geometrical model Fig. 9.

The following areas resulted:

Total Suit	2.83 m^2 (30.56 ft^2)
Both legs	1.12 m^2 (12.05 ft^2)
Both arms	0.57 m^2 (6.10 ft^2)
Maximum projected area	0.62 m^2 (6.71 ft^2)
Minimum projected area	0.23 m^2 (3.06 ft^2)

The effective area available for radiant heat emission from the outer space suit surface would then be:

$$(2.83 - \frac{1.12}{2} - 0.57) = 1.70 \text{ m}^2 \text{ } (18.43 \text{ ft}^2)$$

From present indications it is believed that at a metabolic rate of 2 x 10^6 joules (Approx. 2000 Btu) per hour, a temperature difference between the skin of the man and the outer suit surface of 5^oC (9^oF) or less will be attainable. With a skin temperature of 33^oC (91.4^oF), outer suit surface temperature would be 28^oC, (82.4^oF). Using the thermal emissivity of the Gemini suit helmet (Reference 5, Table 2-8) of 0.88, total emission from the suit will be 2677 x 10^3 joules per hour (2536 Btu per hour). Within the accuracy of the assumptions made, this will be the maximum gross heat emission of the external suit surface. This gross heat emission must balance both, metabolic heat and the heat absorbed from external radiation sources, such as the sun, planetary surfaces and nearby space vehicles.

The amount of radiation absorbed by the suit surface will depend on the absorptivity of the outer suit surface to the wave length of the incident radiation, the intensity of the radiation and the effective area exposed to radiation. The effective area for irradiation from a relative distant object will be equal to the projected area.

The attitude of the astronaut relative to an external radiant source such as the sun, and the temperature of the heat source which determines the wave length of the radiation will be significant factors for the ability of the external suit surface to reject metabolic heat.

In regard to radiant inputs from the sun, advantage will be taken in the usual way of spectral selectivity of the outer suit surface. As an example of the effect of attitude, the solar input for the geometrical model of man, used in this analysis, has been calculated for two (2) different attitudes of an astronaut relative to the sun. In these calculations, a solar absorptivity of 0.25, given in Reference 5 for the Gemini suit helmet was used as being representative for a fiberglass suit shell. The maximum and minimum projected areas were compared. With the maximum projected area facing the sun a heat input of 787×10^3 joules (746 Btu) per hour was calculated. The solar input for the minimum projected area is equal to 360×10^3 joules (341 Btu) per hour.

For an astronaut during EVA, not exposed to planetary radiation or significant radiant inputs from nearby space vehicles, the net metabolic heat rejection by radiation from the outer suit surface will, according to these calculations, vary from approximately 1890×10^3 joules (1790 Btu) per hour to 2317×10^3 (2195 Btu) per hour. This net heat rejection of metabolic heat from the outer suit surface is significant. Weight savings in the order of one kg (2.2 lbs) per hour of EVA may be realized at certain operating conditions and high astronaut activity levels. The heat rejection capability of the outer space suit surface will, however, not suffice under unfavorable conditions, when high metabolic rate and maximum heat absorption from external radiation sources coincide.

Where significant planetary radiation will be incident on the external suit surface as, for example, in the equatorial areas of the moon near lunar noon, heat absorption of the outer suit surface will be very large. Under these conditions, there may be no net heat rejection from the external suit surface possible. This is, of course, due to the fact that radiation from planetary surfaces is in the same part of the spectrum as the emission from the suit and, therefore advantage can not be taken of spectrally selective external suit surfaces. These unfavorable conditions dictate the need to provide an auxillary means of heat rejection. A water evaporating device, either water boiler heat exchanger or sublimation plates, will be required.

This does not diminish the value of the external suit surface as a thermal radiator. Attitude of an astronaut in EVA will change from the unfavorable to more favorable conditions such that, as an average, significant saving of water will be realized during extended time missions.

Even for lunar surface operations, significant savings on water will be realized. The unfavorable condition for radiative heat rejection on the moon is limited to a time period in the order of four (4) to five (5) Earth days, around lunar noon; therefore, during an extended lunar mission the external suit surface will be an effective sink for metabolic heat for about 18 to 20 days of the 28-day lunar cycle.

It is the intent of this research program to integrate means of water evaporation or sublimation into the suit shell such that passive heat transport to auxillary heat sinks is provided and no power penalty for their use is incurred.

Using the same simplified model for suit geometry, the overall thermal conductance required for the extreme cold condition was estimated.

This is the condition where a low metabolic rate and absence of a significant external heat input would coincide, as would be the case for a suited astronaut resting during the lunar night. This value of overall thermal conductance would be a target value which should be obtained when the variable conductance space suit shell provides minimum heat transfer and maximum protection against loss of body heat, i.e., the condition when the vapor transfer valves in the controllable heat pipe are closed and heat transfer is by conduction only.

In order to bring this estimate to the conservative side, it was assumed that all of the external suit surface of 2.83 m^2 (30.56 ft^2) would emit heat by radiation to space. The same assumption as above of a thermal emissivity of 0.88 was used. A metabolic rate for a resting astronaut of 422 000 joules (400 Btu) per hour and a skin temperature of 33°C (91.4°F) were assumed.

From the above assumptions, an overall thermal conductance of 1085 joules/hour - m^2 - °C (.053 Btu/hour - ft^2 - °F) was calculated.

Theory of a Heat Pipe

Considering that the concept selected for passive temperature control of a space suit is based on the type of closed evaporating-condensing cycle with capillary return found in a heat pipe, it is useful to analyize the heat and mass transfer processes taking place in these devices. The type of heat pipe which will be applied in a variable thermal conductance space suit shell will, in geometry and construction significantly differ from heat pipes as they have been studied in the past. This difference is not only due to the heat flow controlling feature The heat pipes used in the variable conductance space suit shell are, when compared to common heat pipes, short and wide with heat transport occuring in three dimensions, i.e., from the skin to the outside face, and in two directions parallel with the surface of the suit. Compared to this, what can be called the "classical" heat pipe, made from a section of pipe or tube, provides essentially one dimensional heat flow. The basic principles of vapor transfer, capillary action, evaporation and condensation apply, however, equally to these devices.

The operational principles of a heat pipe are, therefore, discussed on a "classical" heat pipe as illustrated in Figure 10. This figure shows a pipe with the wick attached as an annulus on the wall. The wick is saturated with a liquid. The material of the wick and the liquid used as working fluid must be such that the liquid wets the wick and will be transported by capillary action of the wick. The pipe is closed on both ends and free of all gases other than the vapor of the working fluid. When heat is put into one end of the apparatus, an increase of vapor pressure of the working fluid will result in evaporation of some of the liquid. The vapor flows to the cold end of the pipe due to the vapor pressure difference resulting from the temperature difference. At the cold end a vapor pressure in excess of the saturation pressure corresponding to the cold end temperature results from the evaporation at the warm end and vapor flow. Condensation of vapor occurs. Heat is given off as a result of condensation. By the processes of liquid evaporation at the warm end, flow of vapor from the the warm end to the cold end as result of vapor pressure difference and condensation at the cold end, heat energy is transported from the warm end to the cold end of the heat pipe.

Evaporation of the liquid at the warm end causes the liquid-vapor interface to retreat into the capillaries, resulting in a contact angle of the liquid within the capillaries which is smaller than in the cold end where liquid is being added as a result of condensation. Liquid is therefore pumped by capillary action from the condenser to the evaporator because of the difference in contact angle in the capillary passages of the wick.

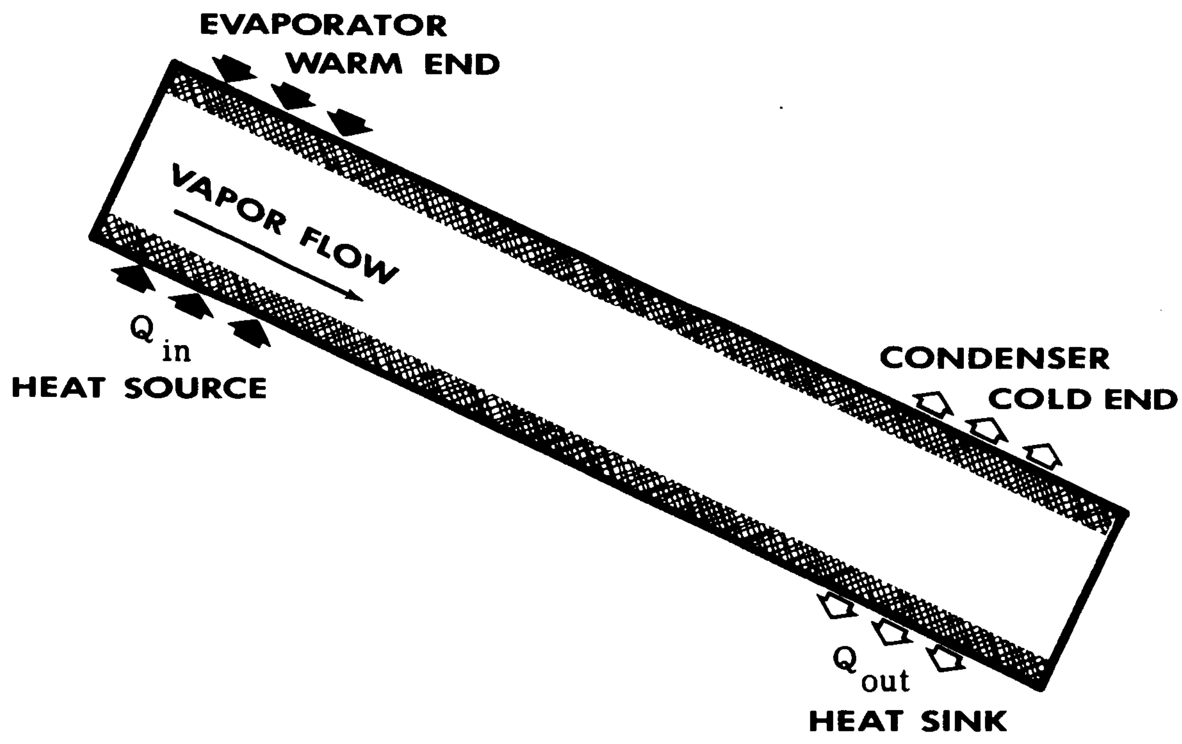


FIG. 10 HEAT PIPE

There are four (4) transport processes occurring simultaneously in a heat pipe.

1. Conductive heat transfer from the outside of the heat pipe to the liquid-vapor interface at the surface of the wick at the warm end, and vaporization of the liquid.

2. Mass transfer of vapor from the warm to the cold end.

3. Condensation and conductive heat transfer at the cold end from the vapor liquid interface at the wick surface to the outside of the heat pipe.

4. Liquid mass transfer from the cold end to the warm end by capillary pumping.

Each of these four processes can be a limiting factor to heat flow performance in a heat pipe. Processes Number 1, 2, and 3 can be considered as three resistances in a series arrangement. These three processes determine the total temperature drop from the outside face of the warm end of the heat pipe to the outside face at the cold end of the heat pipe.

Process 1 and 3 require enough temperature difference to overcome the conductive resistance of the pipe wall and wick plus the amount of super heat, or super cooling, required respectively for evaporation and condensation. In order to keep these gradients at a minimum, high conductivity of heat pipe wall and the liquid-saturated wick are desirable. Therefore, it is desirable to use wick structures of high thermal conductivity and working fluids which are good conductors of heat. Short of being able to achieve this, the wick layer should be made as thin as possible. Good thermal contact, preferably by bonding, between the inner face of the heat pipe wall and the wick is required.

Process Number 2 requires a temperature gradient in order to generate a pressure difference between the warm and cold end of the heat pipe and thereby induce vapor flow. As the vapor flow cross section in most heat pipes is large, the required temperature difference is usually very small. In application to a heat pipe controlled by a valve in the vapor passage, this temperature difference can be controlled by use of a valve with modulating characteristics.

Process Number 4 will in most applications be the limiting factor for maximum possible heat flow in a given heat pipe. It is not, however, a series resistance element in the heat flow path. As long as the wick in a heat pipe is capable of returning an amount of liquid to the warm end of the heat pipe equal to that condensed in the cold end, heat flow will take place. When evaporation and condensing rates exceed the pumping capability of the wick, dry-out of the wick at the warm end will result and heat transport by the heat pipe will stop.

An exception to this occurs when significant pressure differences result from drastic restrictions of the vapor flow passages and capillary pressure becomes unable to overcome a pressure difference existing between the warm and cold end of the heat pipe This will occur in the variable conductance heat pipe which uses a valve in the vapor flow passage. It can be expected that when the vapor flow passage valve closes, a pressure difference between the warm and cold chamber will be generated which will counteract capillary action and lead to dry-out of the wick at the warm (evaporator) end. For application of the heat pipe to a space suit it will be necessary to use wicks which are capable after dry out to rapidly absorb and transport working fluid to the warm end of the heat pipe when the vapor valves open, in order to get fast response times to a control signal demanding increased heat transport.

A second mechanism exists which can cause dry-out at the wick at the warm end of a heat pipe. The temperature at the fluid-vapor interface at the warm end of the heat pipe is equal to the saturation temperature. The liquid within the capillary material at the warm end of the heat pipe and close to the wall of the pipe must, therefore, be super heated relative to the saturation temperature at the liquid vapor interface by a temperature difference equal to the temperature gradient across the wick. If the amount of super heat at this point is large enough to result in vapor formation at the pipe wall, a vapor blanket may be formed which will interfere with conductive heat transfer to the liquid-vapor interface and may drive liquid out of the capillaries of the wick. This phenomenon is unlikely to occur at the rates of heat flow anticipated in a space suit.

An analysis on capillary pumping by a wick is presented in the following. This analysis is similar to analytical treatment presented in the existing literature and essentially based on Reference 6

Capillary Pumping. — Considering the system illustrated in Figure 10, P_{cv} and P_{ev} are the pressures that exist in the vapor space at the condenser and evaporator respectively and $P_{ev} > P_{cv}$. An equation shall be derived for the liquid flow rate in the capillary wick by making the assumptions that the liquid properties are constant, and that the heat pipe is one dimensional. The pressure in the liquid at the condenser is

$$P_{c\ell} = P_{cv} - \frac{p\,\sigma}{\varphi} \cos\theta_c + \rho_\ell g H \cos\beta \qquad (1)$$

where p is the wetted perimeter per unit area of the capillary structure, φ is the wick porosity. θ_c is the contact angle, σ is the liquid surface tension, and β is the angle between the axis of the pipe and the direction of the gravitational field. The pressure in the liquid at the evaporator is

$$P_{e\ell} = P_{ev} - \frac{p\,\sigma}{\varphi} \cos\theta_e \qquad (2)$$

The pressure difference which drives the liquid through the wick material from the condenser to the evaporator is the difference between Equations (1) and (2).

$$P_{c\ell} - P_{e\ell} = (P_{cv} - P_{ev}) + \left(\frac{p\,\sigma}{\varphi}\right)(\cos\theta_e - \cos\theta_c) - \rho_\ell g H \cos\beta \qquad (3)$$

Under steady state operating conditions this pressure difference is opposed by the viscous press loss. The flow in the capillary structure of the wick is laminar and Darcy's law can be used to calculate the frictional pressure drop (Ref. 7). Darcy's law states,

$$P_{c\ell} - P_{e\ell} = \left(\frac{\mu}{k}\right) L \left(\frac{F}{A}\right) \qquad (4)$$

where μ is the liquid viscosity, k is the specific permeability of the wick, F is the volumetric flow rate and A is the cross sectional area of the wick. Substituting equation (4) into Equation (3) gives a relationship for the liquid flow rate from the condenser to the evaporator,

$$F = \frac{Ak}{L}\left(\frac{\sigma}{\mu}\right)\left\{\frac{p}{\varphi}(\cos\theta_e - \cos\theta_c) - \frac{\rho_\ell g H}{\sigma}\cos\beta - \left(\frac{P_{cv} - P_{ev}}{\sigma}\right)\right\} \qquad (5)$$

The pressure drop in the vapor space is for most heat pipes very small relative to other pressure drops in the system. Therefore, Equation (5) may be simplified to give,

$$F = \left(\frac{Ak}{L}\right)\left(\frac{\sigma}{\mu}\right)\left\{\frac{p}{\varphi}(\cos\theta_e - \cos\theta_c) - \frac{\rho_\ell g H}{\sigma}\cos\beta\right\} \qquad (6)$$

As the input power to the heat pipe is increased and the rates of vaporization and condensation increase, the apparent contact angles of the liquid in the wick, θ_e and θ_c, change so that $(\cos\theta_e - \cos\theta_c)$ becomes more positive. This gives a greater capillary driving force and thus a larger liquid flow rate. The maximum flow rate occurs when $(\cos\theta_e - \cos\theta_c) \approx 1$. This means that the contact angle is zero in the evaporator and 90 degrees in the condenser.

Because of the high heat transport rates resulting from the mass transfer and phase change system in a heat pipe, the temperature difference of the liquid over the length of the heat pipe and the sensible heat transferred is very small relative to the latent heat transferred. Therefore, we can assume that all the heat is transferred as latent heat and write

$$Q = \rho_\ell \lambda F \tag{7}$$

where Q is the heat transfer rate and λ is the latent heat of vaporization. Substitution of Equation (7) into (6) gives the following equation for heat transfer rate,

$$Q = \left(\frac{Ak}{L}\right)\left(\frac{\rho_\ell \lambda \sigma}{\mu}\right)\left[\frac{p}{\varphi}(\cos\theta_e - \cos\theta_c) - \frac{\rho_\ell g H}{\sigma}\cos\beta\right] \tag{8}$$

The maximum heat transfer rate that the heat pipe can maintain, assuming that it is liquid flow limited, occurs when the liquid flow rate is maximum. As discussed above, this occurs when $\cos\theta_e - \cos\theta_c = 1$. The maximum heat transfer rate is given by the equation,

$$Q_{max} = \overset{\mathrm{I}}{\left(\frac{Akp}{\varphi L}\right)}\overset{\mathrm{II}}{\left(\frac{\rho_\ell \lambda \sigma}{\mu}\right)}\overset{\mathrm{III}}{\left[1 - \frac{\rho_\ell g H \varphi}{p\sigma}\cos\beta\right]} \tag{9}$$

Equation (9) shows that Q_{max} is the product of three factors: The first (I) is a grouping of wick properties and the second (II) is a grouping of the working fluid properties. In choosing a wick and working fluid, the designer wishes to make these groupings as large as possible. The third factor (III) represents the relative importance of gravitational forces. Under conditions in space where g is essentially equal to zero, this term is unity.

Working Fluid Selection

For the application of a heat pipe to a space suit, the selection of the proper working fluid is of critical importance. It cannot be based on the properties which lead to high performance of a heat pipe alone. Considerations of suitable temperature range, safety and material compatibility are of importance and may be the determining factors.

For a given wick material, a working fluid can be selected for maximum heat transport capability on the basis of group II of equation (9). This group of fluid properties

$$G_{II} = \frac{\rho_{\ell}\lambda\sigma}{\mu}$$

indicates that for a constant contact angle between wick material and working fluid, maximum possible heat transfer in a heat pipe will increase with increasing liquid density ρ_{ℓ}, heat of vaporization λ and surface tension σ, and decrease with increasing viscosity μ. Figure 11 shows a comparison of several heat pipe working fluids based on this fluid property group.

The temperature range of a heat pipe working fluid is defined by its freezing point and by its critical temperature. The range over which a working fluid can operate requires that it must be a liquid at the lowest and at the highest temperature it will experience. Solidification of the liquid phase in a heat pipe would stop heat transfer. The freezing will occur at the cold end of the heat pipe and there will be an eventual transfer of all of the working fluid to the cold end. Drop of temperature at the cold end below the freezing point of the working fluid will be accompanied by a corresponding drop in vapor pressure which will cause evaporation and vapor flow from the wick at the warm end until dry-out at the warm end occurs. Theory and experiment show that start up of a heat pipe after the working fluid has been solidified, is extremely slow. The only mode of heat transfer from the warm end to the cold end of the heat pipe is by conduction in the heat pipe structure.

On the other end of the temperature range, the critical temperature of the working fluid will obviously stop the function of the heat pipe. The critical temperature is that temperature at which a fluid ceases to exist in the liquid phase. Latent heat of evaporation diminishes rapidly on approach to the critical temperature. Efficient

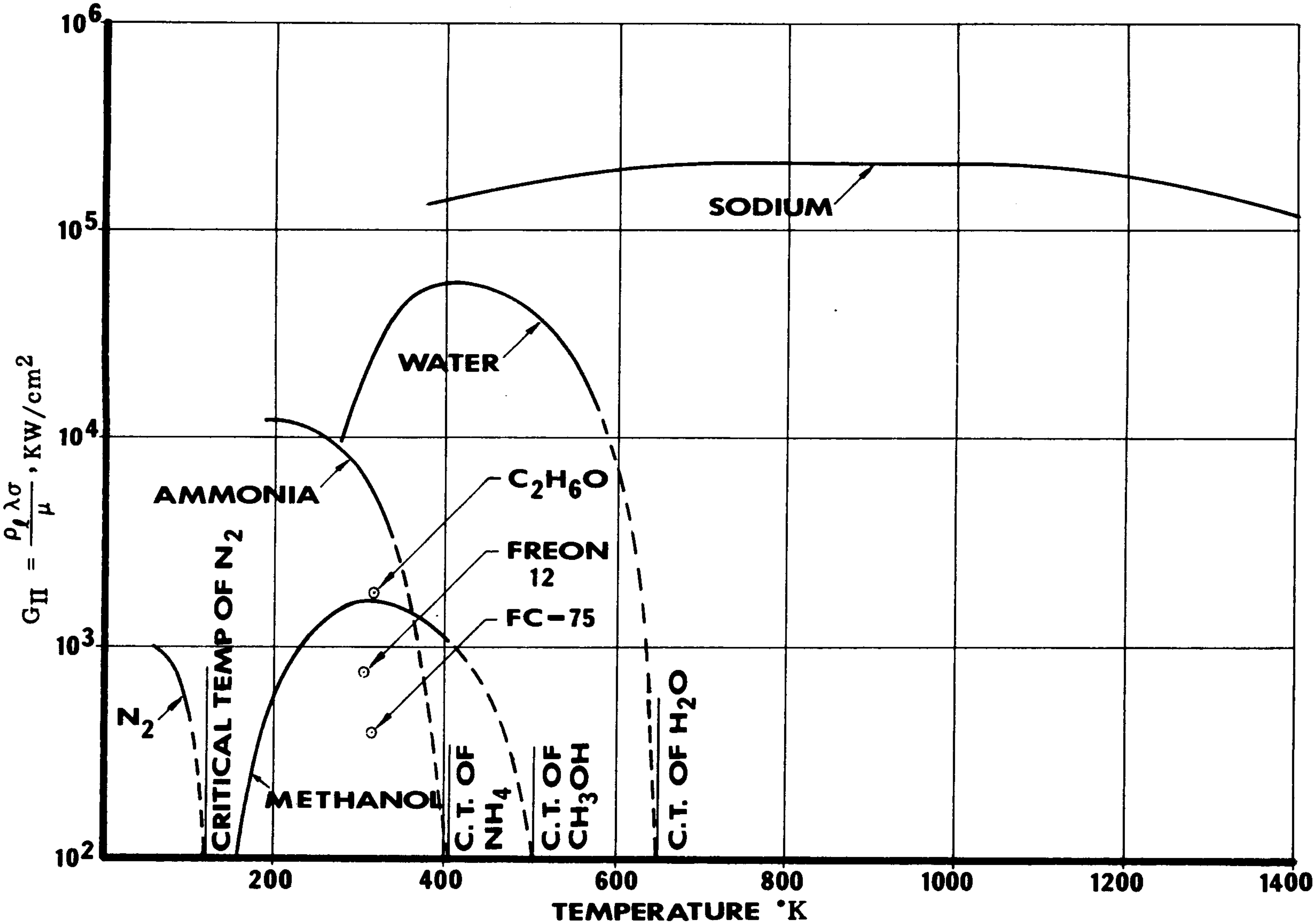


FIG. 11 $G_{II} = \frac{\rho_\ell \lambda \sigma}{\mu}$ VERSUS TEMERATURE FOR A NUMBER OF HEAT PIPE FLUIDS heat pipe operation requires working fluids with a critical temperature well above the maximum operating temperature.

For the temperature range of the human body, the superiority of water as a working fluid is apparent. Unfortunately, it will not be possible to use water alone as working fluid for space suit applications because of its high freezing point. It must be expected that the temperature on the external face of the space suit shell will fall far below the freezing point of water, when low metabolic rate and/or absence of significant external heat inputs occur. For the performance of the present study, it was assumed that methyl alcohol would be used in the controllable external heat pipes of the space suit, while water would be used in the heat pipes of the pressure shell in proximity of the human body. Future investigations will explore the possibility of freeze prevention by use of aqueous solutions.

EXPERIMENTAL RESEARCH

Experimental Research Approach

The theoretical and analytical investigations of passive control of temperature in a space suit by the use of a variable conductance space suit shell verified usefulness and feasibility of the concept. Experimental research was initiated in order to verify theoretical considerations, generate performance data and develop a fabrication technology for passive temperature control for space suits.

In order to be able to perform experimentation verifying the functional and thermodynamic aspects simultaneous with the activities directed towards development of a fabrication technology, it was decided to separate these two areas of research activities. Functional devices which simulated the thermodynamic functions of the variable conductance space suit shell were built of readily available materials. Simple geometries were selected which permitted easy and low cost fabrication. This approach permitted functional experimentation and provided thermodynamic data indepedent of the development of a fabrication technology. Development of the fabrication technology was persued as a parallel effort.

While this approach proved itself as basically useful in being able to obtain satisfactory results in both areas of investigation, the assumption that well-established techniques existed for fabrication of simulation devices proved to be over optimistic. It became apparent that the techniques used for the fabrication of heat pipes of circular cross section were not adequate when the geometry of the devices deviated significantly from that of a pipe.

It was also recognized that it would be desirable to gain a better understanding of the functional aspects of a heat pipe. Ideally, it should be possible to obtain enough experimental data on each of the four basic processes of heat pipe operation, described in the preceding chapter, "Theory of a Heat Pipe", to be able to analytically predict performance of any design and configuration of a heat pipe type device. Such an investigation into the basics of heat pipe operation would, however, have gone far beyond the scope of this program. Some experiments leading to generation of basic data were performed. Others were, however, deleted when it became apparent that testing of a complete simulation device would provide the required results at lower cost.

Wick Evaluation

Of the four transport processes described under "Theory of a Heat Pipe," the mass transfer of the liquid working fluid by capillary action is the least predictable. A wick is a random arrangement of fibers and pores with usually unknown distribution of pore size, direction of capillaries and other characteristics. Wicks considered for space suit heat pipes, may originally have been designed and fabricated for use as filters, or electrical, thermal or accoustic insulation. It was therefore necessary to base the selection of wicks on experimentation with a number of commercially available fibrous and textile materials and to select those which have the most satisfactory performance for the specific application.

Methyl alcohol and water are the two working fluids primarily considered for application in the variable conductance space suit shell. Therefore wick performance data for these two fluids were required. Previous studies performed under Contract NAS2-2102 (Reference 1) have shown that a group of materials, available under the trade name of "Refrasil" has given very good capillary performance with water. These materials are spun, woven or braided from silicon dioxide fibers and available from the Hitco Corporation of Gardena, California. They are manufactured for the purpose of high temperature electrical and thermal insulation. Chemically, they are similar to silica gel, and their outstanding capillary performance with water is probably due to their excellent wetting characteristics.

Because of the availability of data for wick performance with water from Reference 1, the wick-testing program was limited to testing of wicks with methyl alcohol.

The criterion which determines the usefulness of a wick for application to a heat pipe is its ability to transport the liquid working fluid against an imposed head. This head would, under zero "g" conditions, result from the pressure difference which is required to induce flow of vapor from the warm to the cold end of the heat pipe. In most applications, this pressure head will be very small. In an acceleration field, the weight of the working fluid, as it results from the magnitude of the existing "g" forces, (i.e., earth or lunar "g", thruster accelerations) will also have to be overcome by capillary pressure. For the purpose of comparison of various wicks, the capability of the wick to lift a liquid vertically is a good criterion.

Testing of wicks was performed by measuring the rate of lift in vertical wicks, relative to the liquid head in the wick. Liquid weight retained by capillary action in a unit volume of wick was determined. The liquid mass flow rate in the wicks, against liquid heads counteracting the flow, was then calculated.

The experiments were performed by enclosing wicks in a vertical position in glass tubes with the bottom of the wick and glass tube submerged in methyl alcohol (Fig. 12). The wicks were arranged free hanging in the glass tube. A small opening on the top of the tube permitted pressure equalization of the tube with the laboratory atmosphere. This arrangement did provide an atmosphere around the wick essentially saturated with methyl alcohol vapor. This avoided that evaporation from the wick would bias the indicated liquid flow performance of the wick. Liquid rise in the wicks was made visible by putting dots of methyl alcohol-soluble dye (Speedry Chemical Products Inc., "Magic Marker") in regular intervals on the wick. As the liquid rose in the wick, these dots were washed out, giving an indication of the progress of the liquid. This technique is believed to give relatively correct indications of the location of a liquid while it rises up into a wick. Visible observations of the location of a liquid, rising in a wick, is made difficult by the fact that any additive such as a dye, when mixed with the liquid, could change surface tension of the liquid, which would cause significant errors. By putting small dye dots on the wick, the contamination of the fluid to be tested is minimized, although it is recognized that it is not completely avoided.

Time rate of lift was measured with a stopwatch. Figure 13 shows measured lift height versus time. In order to determine flow

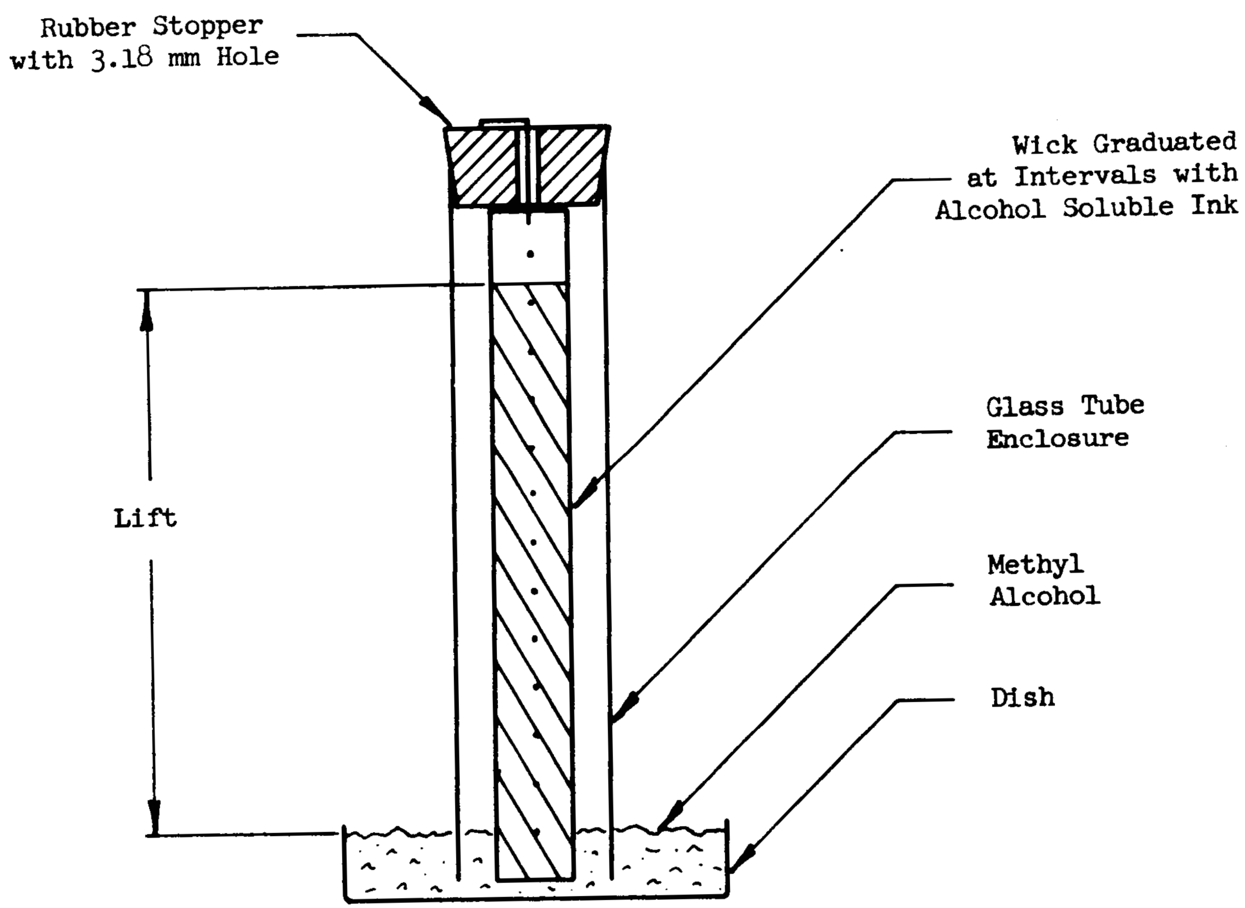


FIG. 12 SCHEMATIC OF TEST SET-UP FOR TESTING WICK LIFT CAPABILITY

FOR SAMPLE IDENTIFICATION: SEE TABLE I

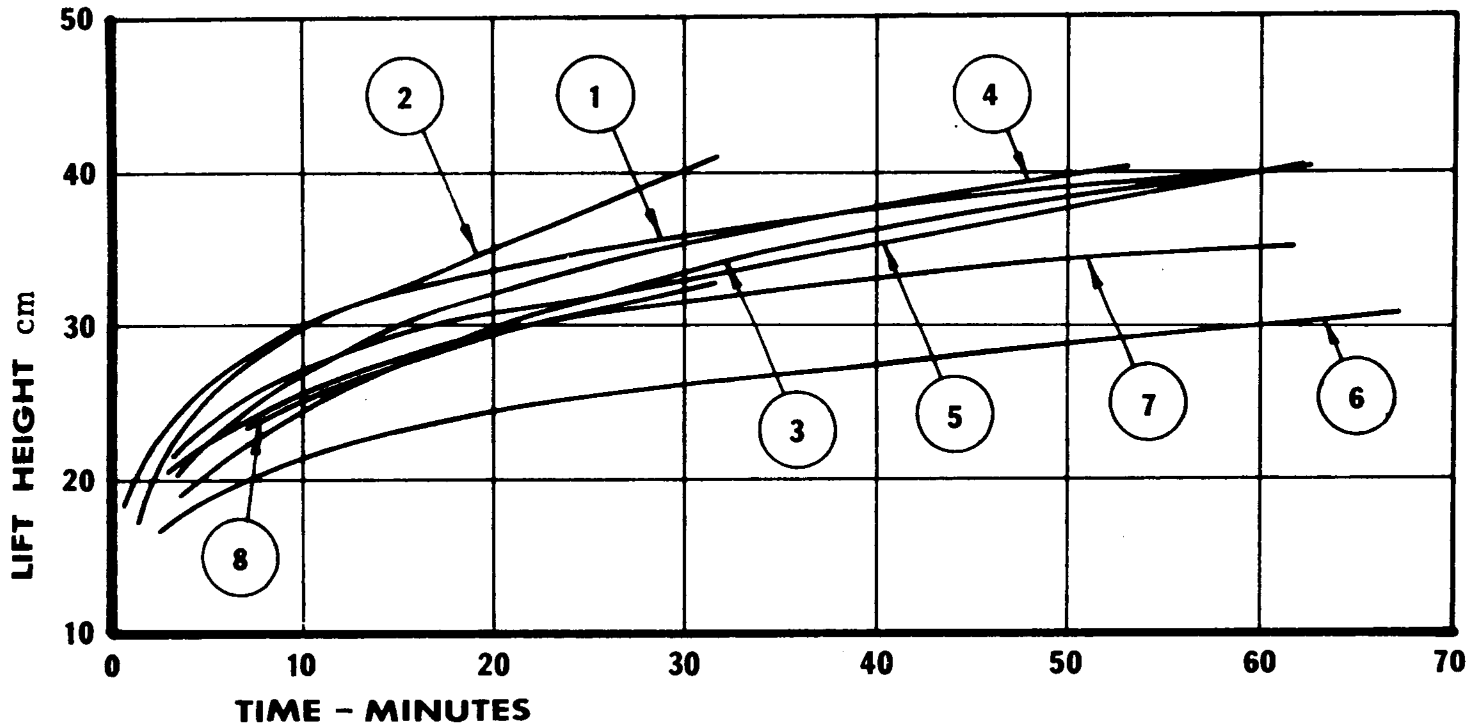


FIG. 13 METHYL ALCOHOL CAPILLARY LIFT HEIGHT IN VARIOUS WICKS AS FUNCTION OF TIME

rate by capillary action as a function of lift height, the liquid volume retained by capillary effect, in the wicks tested, was determined. Samples of the candidate wicks were cut to exact dimensions. The samples were then dipped into a graduated cylinder filled with methyl alcohol until fully soaked and withdrawn. They were permitted to drip drain for 30 seconds into the graduated cylinder. The amount of liquid removed by the wick from the graduated cylinder was recorded. The data so obtained were then converted to gram of methyl alcohol per cubic centimeter of wick volume.

The advantage of volume measuring is that it can be performed quickly and that evaporating errors are minimized. Comparison of weight of dry and saturated wicks was attempted, but found unsatisfactory for a relatively volatile fluid such as methyl alcohol, because of the slowness of the weighing process. Results of the liquid retention tests are shown in Table I. This table also identifies the samples tested and the source of procurement.

With data on lift height as function of time and methyl alcohol retention weight per unit wick volume available, the methyl alcohol capillary mass flow rate per unit wick area, for various lift heights, was calculated. The weight of liquid lifted by the wick per unit time and unit cross-sectional area

$$W = \frac{\Delta L}{\Delta t} \rho'$$

where: $\frac{\Delta L}{\Delta t} = \frac{\text{lift}}{\text{time}}$

ρ' = weight of liquid retained in a unit volume of wick (from Table I)

The resulting methyl alcohol flow rates as function of lift in the wicks are plotted in Figure 14.

It is recognized that this technique of calculating lift rate also includes a potential source of error and should be considered only as a reasonable approximation. The error results from the fact that saturation of the capillaries of a wick will not be uniform, but will diminish as lift head increases. In a vertical wick, a smaller amount of liquid will be contained near the top, in a unit volume, than in the bottom of the wick. This will be due to the fact that, as the lift height increases, only the smaller capillaries available in a wick will contain liquid because the maximum lifting capability of the larger capillaries has been exceeded.

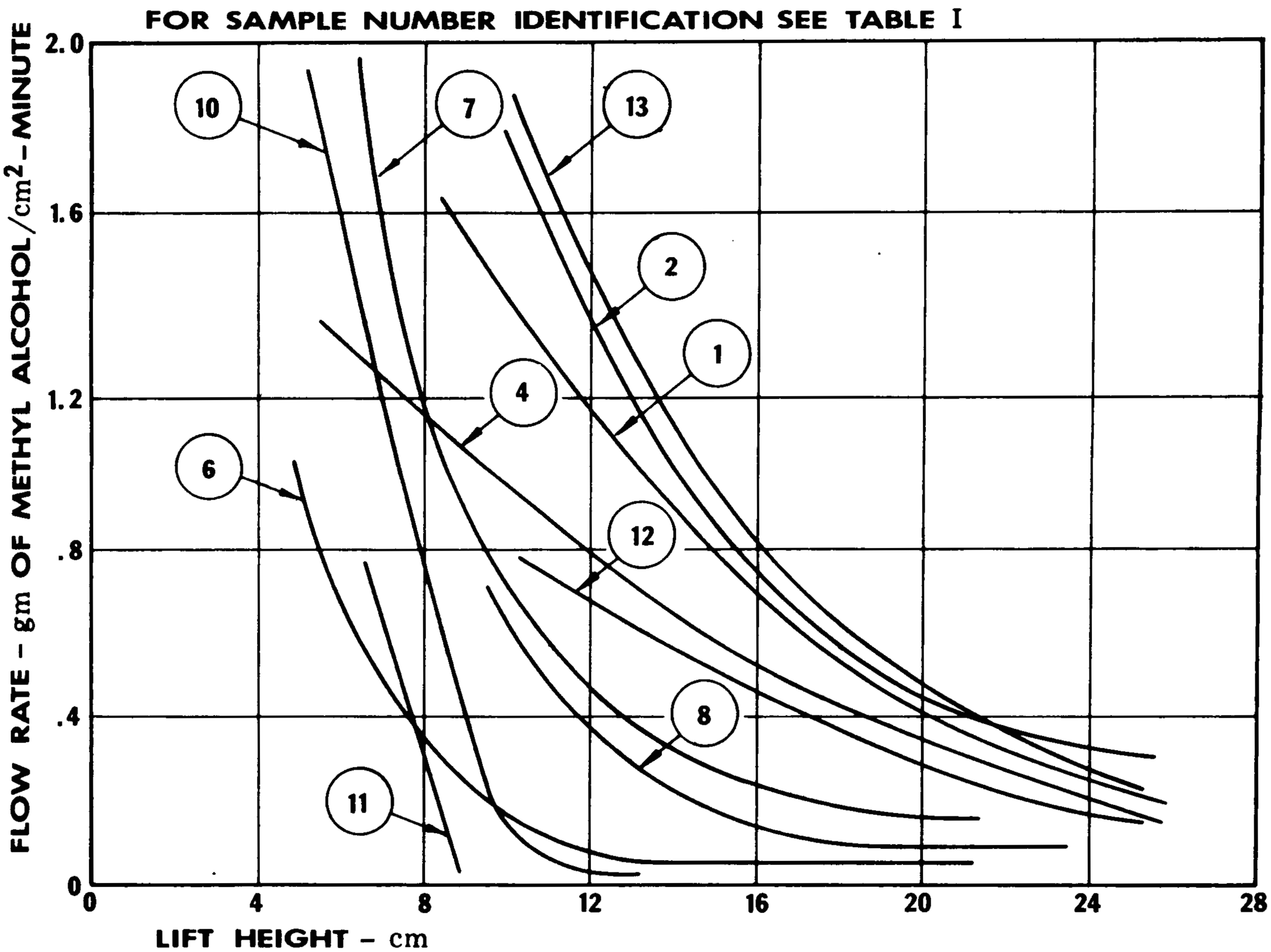


FIG. 14 METHYL ALCOHOL CAPILLARY FLOW RATE IN VARIOUS WICKS AS FUNCTION OF LIFT

TABLE I
SIZE & SOURCE OF WICK MATERIALS TESTED FOR
METHYL ALCOHOL LIFT CAPABILITY

Wick Sample No.	Wick Description		ρ' gm Methyl Alcohol / cc of Wick	As Received Condition Sample Size cm	
	Company	Designation		Cross-Section	Length
1	Hitco Gardena, Calif.	Refrasil Sr. N .953 cm Sleeving	.69	.953 x .079	30.48
2	" "	Refrasil Sr. B 2.54 cm Sleeving	.66	2.54 x .24	30.48
3	" "	Refrasil Sr. B .16 cm Sleeving	.48	.16 x .12	30.48
4	" "	Refrasil Sr. SS .16 cm Sleeving	.37	.16 x .24	30.48
5	" "	Refrasil Sr. S-100-3 .953 cm Sleeving	.21	.953 x .12	30.48
6	" "	Refrasil Sr. T-100-1 .254 cm Tape	.43	.254 x .04	30.48
7	" "	Refrasil Sr. C-100-48 Cloth	.47	.254 x .08	25.40
8	American Felt Co. Glenville, Conn.	Dacron 65-DA22-1	.28	1.90 x .32	24.45
9	" "	Nylon 61-NY18-5	.61	1.90 x .24	25.40
10	" "	Nylon, NY 18-1	.66	1.90 x .19	25.40
11	" "	Polypropylene PO-7008	.75	1.90 x .24	25.40
12	Atlas Asbestoes N. Wales, Penn.	2.54 cm Glasweb Tape Style 2040	.58	2.54 x .32	30.48
13	" "	3.81 cm Glasweb Tape Style 1981	.68	3.81 x .16	30.48

These tests are, however, still considered valid for the purpose for which they were performed. Purpose of these tests was selection of the wick with the best transport capability out of a number of potential materials. They were, therefore, comparative rather than absolute. Errors in the observation, as they have been recognized as unavoidable, will be essentially equal for all specimen and the rank order of lifting capability will still be suitable for selection of a wick.

As stated above, previous work performed under Contract NAS2-2102 has shown Refrasil to provide optimum performance for water (Reference 1). In the tests with methyl alcohol, it was found that the same wicks also provided maximum performance of flow versus head with methyl alcohol. This result could not necessarily be anticipated. In-house technology research performed earlier by TRW Systems had, for example, indicated the same wick to be a rather poor performer with Freon 11.

For application of heat pipes to space suit thermal control, using water and methyl alcohol as working fluids, Refrasil wicks were selected on the basis of the results of these investigations.

Evaporation From Wick Surfaces

Evaporation from the surface of a wick is one of the basic processes on which the function of a heat pipe depends. The purpose of the tests, described in the following, was therefore to observe and measure evaporation rates of water and methyl alcohol from the surface of a liquid-soaked Refrasil wick. There were good reasons for interest in this particular phenomenon.

The fluid property group $\frac{\rho_l \lambda \sigma}{\mu}$ for evaluation of working fluids includes only properties which would effect transport efficiency of a wick with the particular fluid. The fluid property group is, therefore, a performance criterion for heat pipes when the capability of the wick to return the liquid phase of the working fluid from the cold to the warm end is the limiting factor. The fluid property group does not relate to the rate of evaporation of the working fluid. It is, however, the product of evaporation rate and heat of vaporization which will determine heat flow rate attainable for a given temperature gradient, when a heat pipe operates below the limitations of wick performance.

Furthermore, the highest temperature in the liquid-filled matrix of the wick will exist at the side of the wick which is bonded to the heat pipe wall. Vapor pressure of the liquid in the wick should, therefore, be highest at the heat pipe wall-wick interface. Should this vapor pressure result in boiling at the interface, the vapor formed would counteract the capillary forces of the wick and drive the liquid out of the capillaries. This would result in wick dry-out and break down of heat pipe activity. Heat pipes do however in fact operate. It must be therefore concluded that this effect does not take place at heat flow rates at which heat pipes operate. It has been theorized that, as a result of capillary pressure, the pressure at the heat pipe wall-wick interface is higher than at the liquid-vapor interface.

The experimentation performed was not of a highly sophisticated nature. It was realized that an extensive study of wick surface boiling phenomena would be beyond the scope of this program. A simple apparatus was designed and built and limited observations were made, which confirmed that wick dry-out of the type described above would not occur at heat fluxes anticipated for a space suit. Further indications obtained were that the relative performance of water and methyl alcohol depended on temperature. Water appears to be a more efficient working fluid at human body temperatures. At temperatures as they may occur near or at the external suit surface, methyl alcohol seems to be equal or better than water in rate of evaporation.

Experimental Apparatus.- An experimental apparatus was designed and fabricated for the performance of these tests. It consisted of a cylindrical chamber made from a circular Plexiglass ring with an aluminum bottom and a Plexiglass top, as shown in Figure 15. The aluminum bottom was machined to a flat raised center section, surrounded by a trough. The wick to be tested was bonded to the raised center section and extended down into the trough. Thermocouples were installed on the top of the wick and at the interface between the aluminum plate and the wick. Connections were provided in the top plate for a vacuum pump, an absolute pressure gauge and a liquid supply from a titration tube. The connections to the vacuum pump permitted evacuation of the internal cavity of the cylinder. A cold trap in an ice bath was arranged between the test fixture and the vacuum pump for condensation of vapor. An infrared lamp supplied radiant heat into the bottom of the aluminum plate. A "Variac" was used as power supply to the heat lamp to permit control of heat input.

Test Procedure.- The following procedure was followed in performing the experiments:

- The chamber was evacuated to a hard vacuum (approximately 10^{-6} torr).
- Liquid was supplied to the chamber from the titration tube until the wick was saturated and the trough in the bottom of

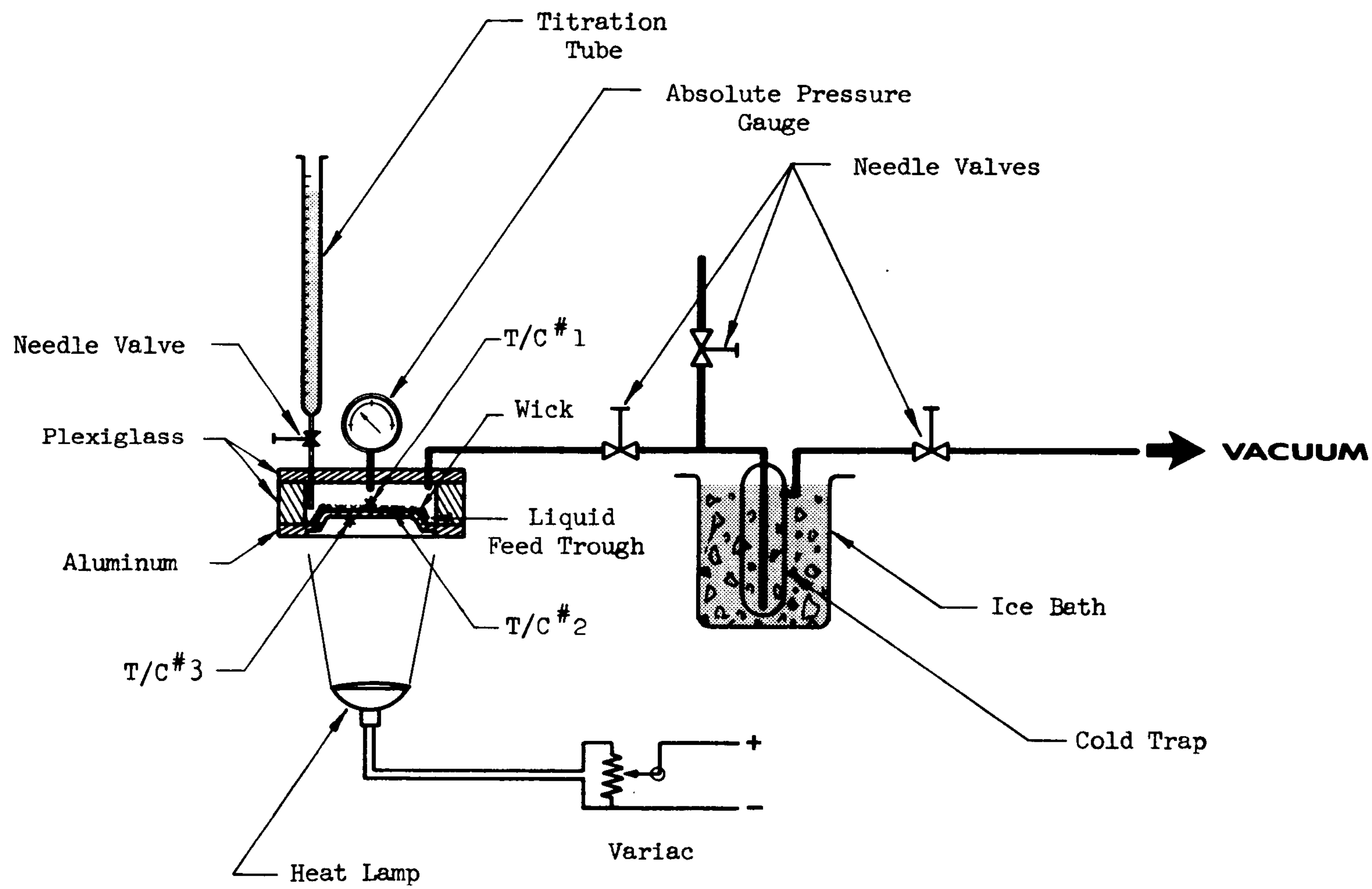


FIG. 15 TEST ARRANGEMENT, MEASUREMENT OF EVAPORATION RATE FROM A WICK SURFACE

the chamber was nearly filled with the liquid. The heat lamp was turned on, and after an initial warm-up time, the valves regulating the pressure chamber were adjusted to obtain the desired pressure.

- Liquid level in the trough was maintained at a constant level by adjustment of the valve in the bottom of the titration tube and visual observation through the Plexiglass sides and top of the test chamber.

- After steady state operating conditions of pressure and temperature were obtained, which required usually about 1 to $1\frac{1}{2}$ hours of adjustments, thermocouple temperature signals and chamber pressure were recorded. Liquid volume supplied to the chamber, representing the rate of evaporation, was recorded as a function of time.

Interpretation of Test Results.- The data for evaporation rate as a function of chamber temperature and corresponding saturation pressure were used to calculate heat rates for evaporation from the wick surface in terms of heat energy per unit area per unit time. A number of tests points so obtained were plotted as function of liquid-vapor interface temperature, for water and methyl alcohol, in Figure 16. As the figure shows, there was considerable scatter in the test points. This is not too surprising considering that liquid supply was measured on the basis of visual observation of the liquid level in the trough and other shortcomings inherent in a simplified test set up. Averaging lines drawn through the test points indicate that up to a temperature of approximately 17°C, methyl alcohol has higher evaporating rates than water and could be expected to give equal or better performance in a heat pipe than water. At higher temperatures than 17°C, water is superior.

Wick Bonding

It is apparent that one of the resistances to heat flow in a heat pipe-type device is the thermal resistance of the interface between the inside surface of the heat pipe wall and the wick attached to it. Any gap between wick and wall will increase this thermal resistance. For this reason and because it is necessary to retain the wick at the location required by the design of the device, wicks should be bonded to the inner surfaces of the heat pipes.

Bonding of a capillary structure such as a wick will, however, create a problem. Any liquid adhesive brought in contact with the

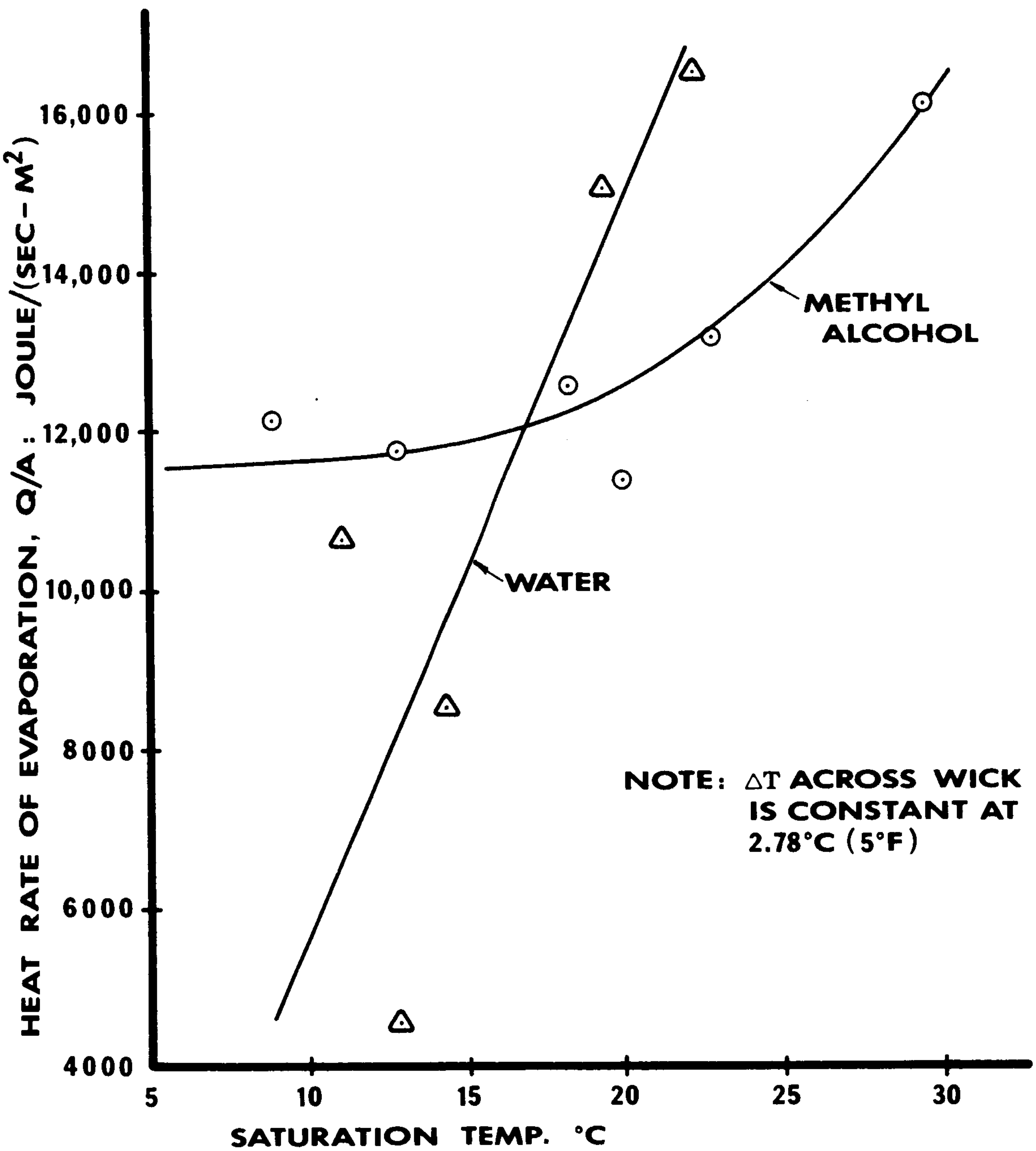


FIG. 16 HEAT RATE OF EVAPORATION VERSUS SATURATION TEMPERATURE FOR WATER AND METHYL ALCOHOL

wick will be drawn by capillary action into the capillaries of the wick and effectively destroy capillary performance. A variety of techniques to avoid this were taken under consideration. One technique would fill the capillaries of the wick with a readily soluble substance prior to bonding it, perform the bonding and then wash the filler material out of the capillaries by a suitable solvent. Another technique would limit the amount of the adhesive such that only a very thin layer of the overall wick thickness would be penetrated.

In the performance of the tasks described in this report, the latter technique was applied. Difficulties were however experienced and extensive experimentation was performed for development of a successful bonding technique and for verification of the reliability of potential techniques over extended time periods at elevated temperatures. These research activities are described in a separate report prepared under the same contract and entitled, "Material Research Report-First Year" TRW SYSTEMS No. 06462-6003-R000, September 1967.

It was found that the best way to limit the amount of adhesive applied to the wick was to use adhesives which were suitable for heat sealing, rather than those relying on solvent evaporation or curing processes.

A technique successfully used at this time takes advantage of a commercially available mylar sheet which is coated on both sides with a polyester resin and available under the name of Schjeldahl No. GT-401. This technique has successfully withstood extended time tests which included exposure to water as working fluid for two weeks at 71°C (160°F.)

The same technique is, however, not satisfactory for use with methyl alcohol at elevated temperatures for extended time periods. During tests, the bonding failed and the wick separated after two weeks of exposure to 71°C (160°F) in the presence of methyl alcohol.

It may be difficult to find a technique which will satisfactory bond Refrasil wicks to fiberglass substrates in the presence of methyl alcohol. Future research will therefore attack the problem in two ways. In addition to research for a suitable bonding technique for Refrasil, consideration will be given to the use of new type of metallic felt wick, which is sealed with a continuous metal sheet on one side. This would, for purposes of bonding, provide protection against penetration of an adhesive into the capillaries of the wick.

Simulation of Space Suit Shell Functions

It was anticipated that the development of fabrication techniques for the fiberglass space suit shell would be a time-consuming process. It was therefore desirable to parallel this effort by a series of experiments which would confirm the functional aspects on which the concept of the variable thermal conductance space suit shell was based.

This space suit shell includes two major functional components:

- The basic structural and pressure shell of the hard space suit, modified into a high thermal conductance structure by converting every internal cavity of this shell into a heat pipe.

- The variable thermal conductance device which would be a controllable heat pipe, using the vapor passage control principle discussed in the introduction.

In both these areas, simulation devices were constructed of readily available and easy to handle materials, such as Plexiglass, aluminum and other metals. The experimentation in these two areas was performed separately, with a high thermal conductance pressure shell simulator and a simulator for the variable thermal conductance suit shell.

Pressure Shell Simulation. - The high thermal conductance pressure shell, when seen from the viewpoint of transversal heat transfer, is a short and wide heat pipe. In order to verify its functional and operational aspects, a simulator was designed and constructed as shown in Figure 17. This simulator was circular and consisted of a short ring of Plexiglass with an aluminum top and bottom plate for heat input and heat removal. Thus, a cylindrical cavity, approximately 6 mm ($\frac{1}{4}$ in.) high was formed.

The two aluminum plates were lined on the inside with sheets of Refrasil wick, which were interconnected by two rolled up pieces of Refrasil wick located in two places between the wick-lined aluminum sheets. A Plexiglass spacer was provided in the center of the two aluminum sheets to increase resistance against collapsing, when the cavity was evacuated. The whole assembly was bonded together with epoxy. A small diameter copper tube was provided from the outside into the cavity for evacuation of noncondensible gases and supplying of water as working fluid. The assembly was mounted on top of a copper block heat sink which could be cooled with chilled water. High thermal

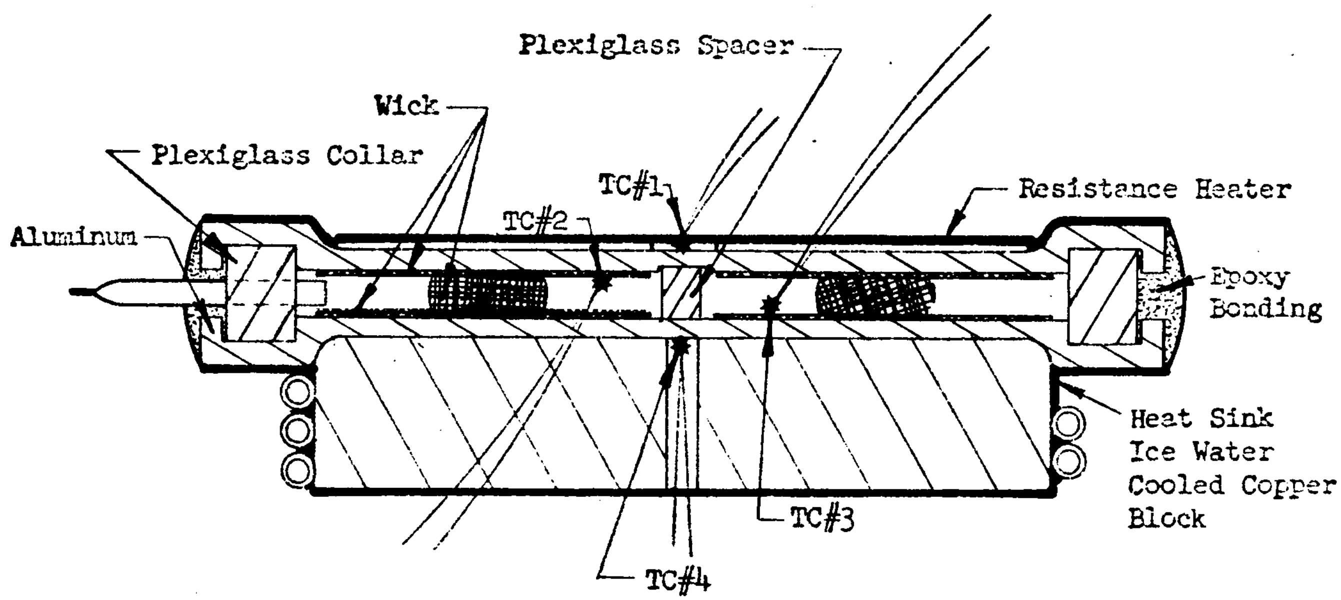


FIG.17 SUIT PANEL SIMULATOR HEAT PIPE

conductivity silicon grease, Dow Corning No. 340, was used to improve heat transfer between the copper heat sink block and the aluminum plate. The opposing aluminum plate of the device was used for heat input. For this purpose an electric resistance heater was bonded to the outside of this aluminum plate. Thermocouples were installed on the outside surfaces of the aluminum plates and internally in contact with the surfaces of the two wicks.

The device was first installed in a vacuum bell jar, with the copper tube open. It was insulated with 20 layers of crinkled aluminum foil. The bell jar was then evacuated to approximately 10^{-6} torr. Power at three different levels as shown on Figure 18 was then applied to the heater and ice water was circulated through the tubes of the heat sink block. Temperature readings of the thermocouples were recorded after the system stabilized in a steady state condition.

The device was then removed from the bell jar and evacuated through the copper tube to a vacuum of approximately 10^{-6} torr. Degassed distilled water was then admitted from a graduated titration tube. The amount of water added had been previously determined as the amount required to completely soak all the wicks present in the cavity.

The copper tube was then sealed off with a pinching tool, the device insulated as before and installed in the vacuum bell jar. The bell jar was evacuated to approximately 10^{-6} torr. The same levels of electric power were again applied to the heater, ice water was circulated through the heat sink block and the system permitted to stabilize. Thermocouple readings were again recorded. Figure 18 shows a plot of temperature at the two (2) thermocouples located at the outside surfaces of the aluminum plates vs. power input.

No reliable readings could be obtained from the thermocouples in contact with the wick surfaces. Readings from these thermocouples were erratic and it was later concluded that the type of installation of these thermocouples caused irregular readings, because the thermocouple wire leads of the thermocouple located on the cold wick surface were in contact with the surface of the warm wick and vice versa. As a result, the thermocouple readings were influenced by thermal conductance in the thermocouple leads and their readings had to be discounted. Comparison of the temperature gradients for equal power input, of the device empty and providing heat transfer only by conduction and radiation, with the much smaller temperature gradients when the device functioned as a heat pipe gave an indication that the modification of the fiberglass pressure shell cavities into heat pipes would provide significant improvement in overall thermal conductance.

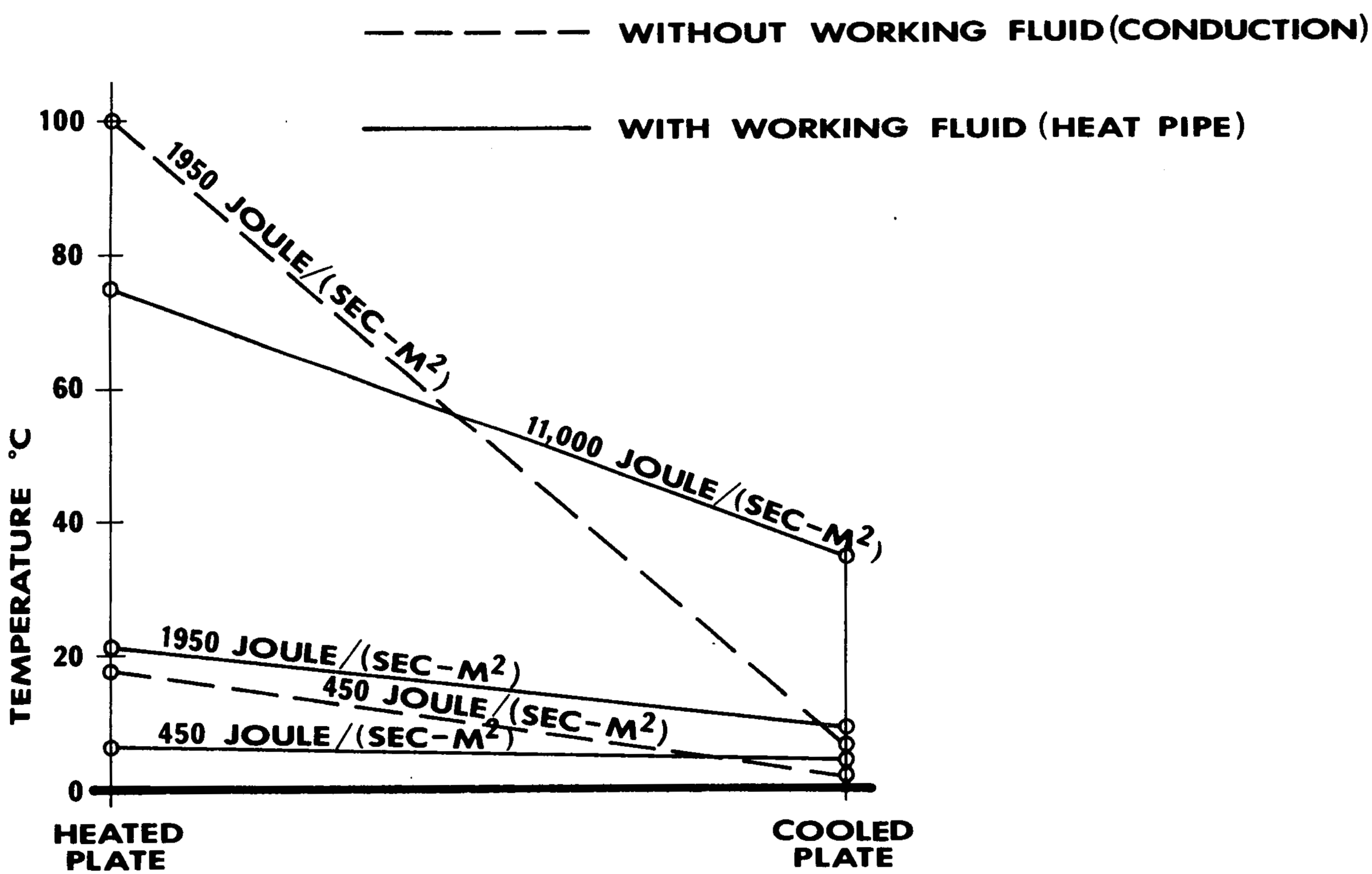


FIG. 18 TEMPERATURE GRADIENT ACROSS SUIT PANEL SIMULATOR

Variable Conductance Space Suit Shell Simulation.- The variable conductance space suit shell is essentially a short, wide heat pipe separated into two chambers. The separating wall has openings for vapor passage and for condensate transfer wicks. The vapor passage is opened or closed as desired to control the amount of heat transported by the device.

The first simulator designed and fabricated was made of a one-inch long piece of four-inch diameter stainless steel tubing with stainless steel cover plates at each end and a stainless steel plate as the chamber separator. The chamber separator had two holes drilled in it. One was for the vapor valve and the other was for a roll of Refrasil wick for condensate transfer. The vapor valve was made from a magnetic type of stainless steel and could be lifted from its seat from the outside of the device by a permanent magnet.

Refrasil sheets were bonded to the inside surfaces of the two cover plates. Carter's rubber cement was used to bond the Refrasil sheets to the stainless steel plates. The rubber cement was applied to the stainless steel surfaces and permitted to dry. The Refrasil wicks were then applied by hand pressure to the dry, but tacky rubber cement. The rationale for the use of this technique was, that absence of a liquid adhesive would protect the capillaries of the wick from being soaked with the adhesive.

The device was assembled by soldering with tin solder, which was later coated with epoxy to cover some small leaks. A small diameter copper tube penetrated into the cavity of the device for evacuation of noncondensible gases and for supplying water as the working fluid. Thermocouples were attached to the outside of each of the cover plates. An electric resistance heater was bonded to the outside of the top cover plate. The device was evacuated and supplied with degassed distilled water in sufficient quantity to saturate all internal Refrasil wicks.

The device was tested under room conditions with power supplied to the resistance heater and heat removed by placing the bottom plate in a dish of ice water. No significant change in the temperature difference across the plates occurred when the vapor valve was opened or closed. It was suspected and eventually confirmed that the wick had become contaminated by the rubber cement when heat was supplied to the top plate. This contamination stopped the capillary action in the wick and the device could no longer function as a heat pipe.

It was then decided that the design of the simulator should be modified to permit disassembly and reassembly of the device in order to make internal modifications possible, when required. A different kind of wick-bonding technique was obviously necessary. As regular rubber cement used in the previous attempt broke down under elevated temperature, it seemed logical to use a similar compound with a higher temperature limit for this purpose. A silicon rubber, Dow Corning RTV-140, was selected for this purpose.

A simulator was designed, made of Plexiglass and aluminum, as shown in Figure 19. It consisted of a Plexiglass body which provided the cylindrical sidewalls and the separating wall dividing the cavity into two chambers. Two aluminum plates closed off the top and bottom. Two Viton O-rings were used to seal the aluminum plates against the Plexiglass body. The device was assembled by the use of two Plexiglass retaining rings and 24 retaining bolts.

The dividing wall contained two holes. One could be opened and closed by a steel valve which could be operated from the outside by a permanent magnet. A short, stainless steel tube was bonded into the other hole through which a roll of wick interconnected the two flat wicks on the inside faces of the two aluminum plates. The device was instrumented with thermocouples on the outside of the aluminum plates and at the surfaces of the wicks. It was installed on the top of a copper block heat sink, which had tubes soldered to it, for chilled water circulation. An evacuation and fill tube connected into the lower of the two chambers of the device. An electric resistance heater was bonded to the top surface of one of the two aluminum plates.

Tests performed with this device were again completely unsuccessful. No significant difference in temperature difference was achieved, whether the vapor transfer valve was open or closed. When tests were consistently unsuccessful, the device was disassembled. It was found that the wicks which had been bonded with silicon rubber RTV-140, had become completely water-repellant. These wicks had been tested and performed perfectly prior to assembly of the device. It was concluded that the RTV-140 had released low-molecular weight silicon oils which had been absorbed into the capillaries of the wicks.

It was at this point in time that it became apparent that bonding of wicks to a solid substrate was a major problem for the development of a variable conductance space suit shell incorporating heat pipes. The decision was made to discontinue for a limited time the efforts of developing the variable thermal conductance simulator and to explore a variety of wick-bonding techniques and their behavior in the presence of low-pressure water or methyl alcohol, in the vapor and liquid phase and with exclusion of noncondensible gases. These

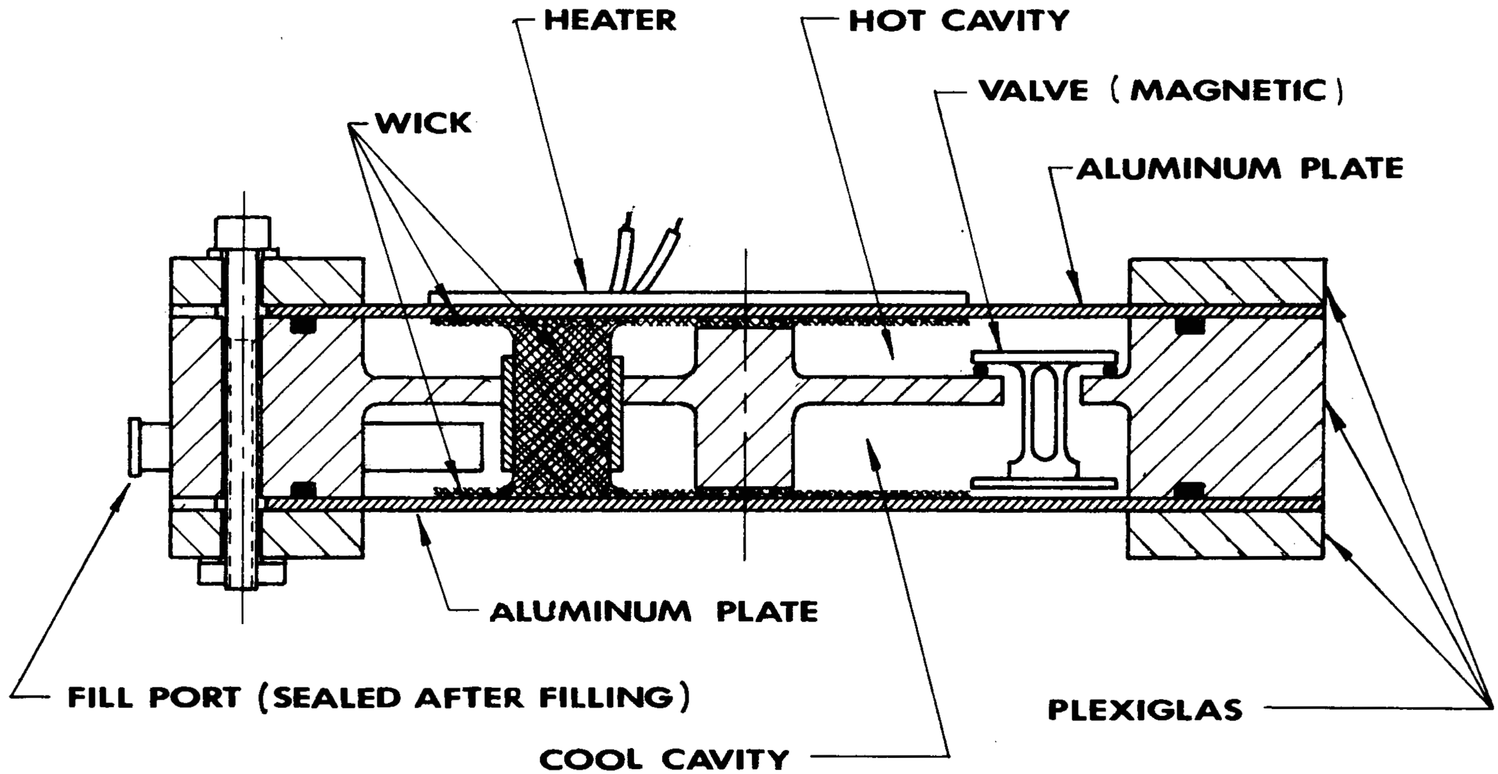


FIG. 19 VARIABLE CONDUCTANCE SPACE SUIT SHELL SIMULATOR

tests were performed at room temperature and elevated temperatures. A report on this phase of the program appears in the "Material Research Report-First Year," TRW Systems No. 06462-6003-R000, September 1967, prepared under the same contract.

After an extended period of research activities in techniques of bonding wicks to solid substrates, one technique was found to be adequate for the durations and temperature ranges of these tests. On the basis of experience with the previous device the design of the simulation device was further modified to the configuration shown in Figure 20.

The new device used two vapor valves in order to be able to study the effect of the cross section of the vapor flow passage on heat transport. The valves were pneumatically remote operated, which permitted placing the whole assembly into a vacuum chamber and use of reflective-type vacuum insulation to approach an adiabatic condition. Heat sealing with polyester-coated mylar sheets as described in the chapter on Wick Bonding, was used to bond the wicks to the metal plates. The heat input plate was made of copper to permit soldering on of the bellows chambers while, for the heat rejection plate, aluminum was maintained as material. Figure 21 is a photograph of the major components of the device prior to assembly.

The device was instrumented with thermocouples as shown in Figure 20. The thermocouples No. 2 and No. 5, located on the heat input and heat removal plate were attached by mechanical peening after the junction was soldered. Thermocouples No. 3 and No. 4 were intended to read the temperature at the liquid-vapor interface on the surface of the wicks. This required that the junctions be held firmly against the surfaces of the two wicks and that other sources of heat flow to the thermocouples be eliminated as far as possible. It is recalled that the techniques used for attachment of the thermocouples to wick surfaces on the tests for "Evaporation From Wick Surfaces" had been unsuccessful. The thermocouples on the new device were supported by the Plexiglass body. Holes were drilled into the side of the Plexiglass body and stiff, 16 gauge copper and Constantan wire was fitted into these holes and sealed with "Torrseal" (made by Varian Assoc.). The wires were brought together with a V-shaped junction and bent in such a way that the junction and approximately 5 mm of the leads were pressed against the wick surface by spring action of the wires. The 5 mm extra contact of the wire leads with the wick surfaces provided improved heat transfer from the liquid-vapor interface on the wick and reduced errors resulting from thermocouple lead heat conductance. Care was taken that thermocouples No. 2 and 5 would not be in contact with the heater or the heat sink block. Leads were kept away, or thermally

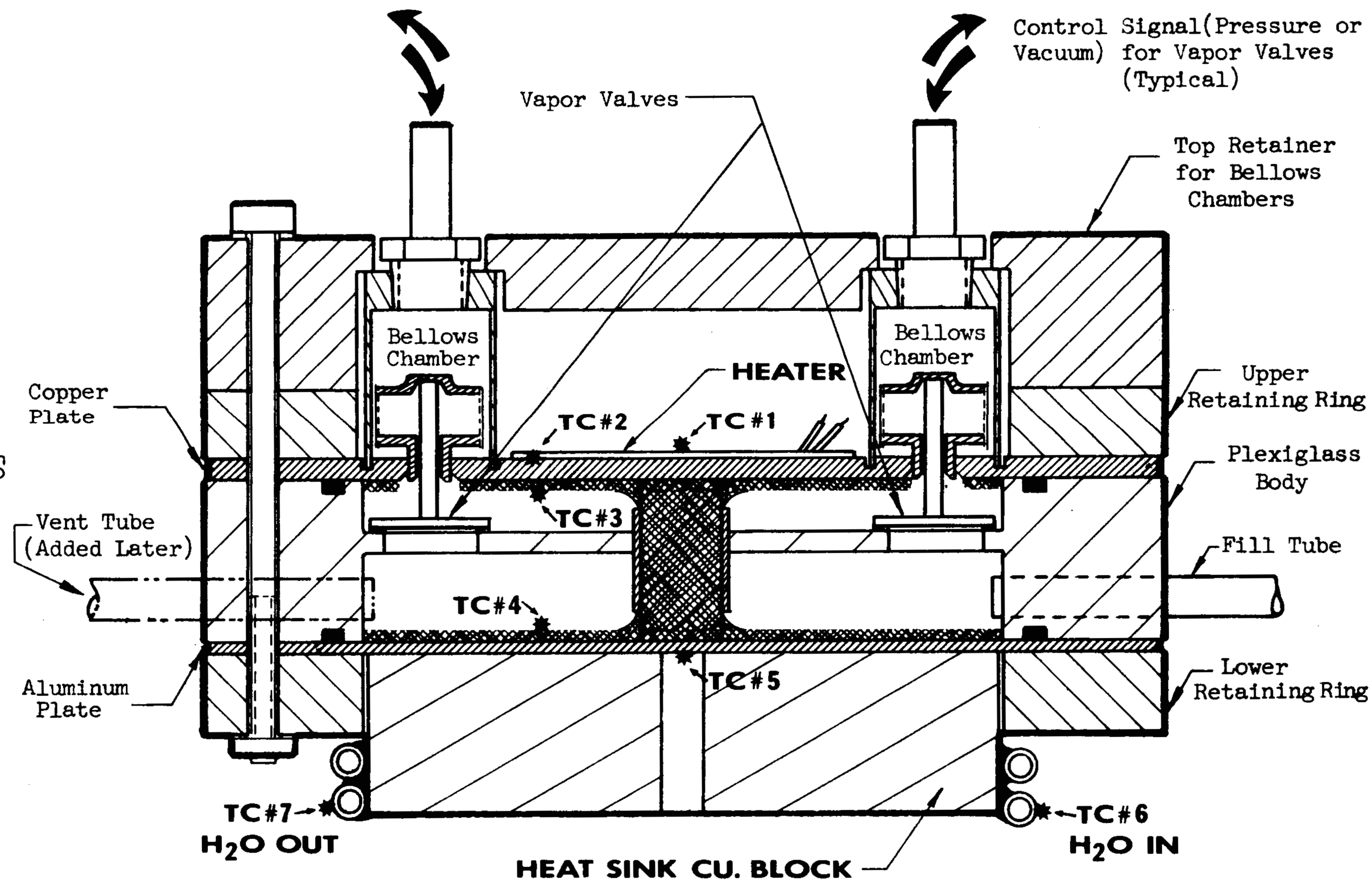


FIG. 20 VARIABLE CONDUCTANCE SPACE SUIT SHELL SIMULATOR (IMPROVED VERSION, SCHEMATIC)

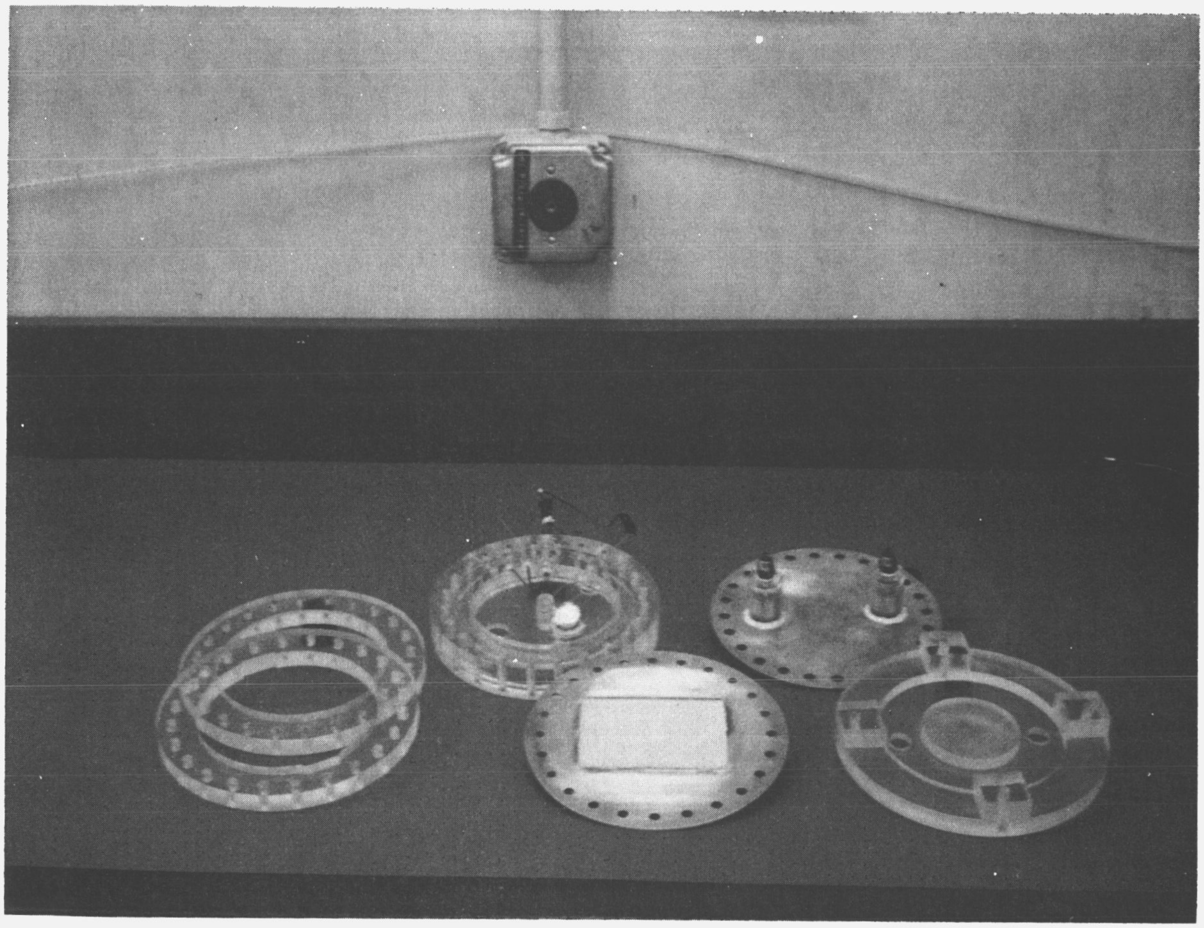


Figure 21 COMPONENTS OF VARIABLE THERMAL CONDUCTANCE SPACE SUIT SHELL SIMULATOR

insulated from external heat sources or heat sinks.

Thermocouple No. 1, located directly on the heater served only for the purpose to monitor the heater temperature in order to protect against damage by overheating. Another thermocouple (not shown) was installed near the outside face of the aluminum foil insulation facing the heater, to provide an indication of the effectiveness of the thermal insulation.

The device was prepared for evacuation and filling by putting a tee onto the fill tube and connecting one branch of the tee to a diffusion pump and the other branch to a titration tube. Valves were installed such that flow could be routed from the interior of the device to the diffusion pump or that water from the titration tube would flow into the device. The device was evacuated and outgassed and distilled water was then supplied in enough quantity to saturate all the wicks. The fill and evacuation tube was then closed off with a pinching tool.

The copper block with the soldered on tubes for circulation of chilled water was used as the heat sink. High thermal conductivity silicon grease, Dow Corning No. 340, provided improved heat transfer between the heat sink block and the heat rejection surface of the device. The device was installed in the vacuum bell jar, insulated with 20 layers of aluminum foil and the bell jar evacuated to 10^{-6} torr. Various power inputs were provided to the electric heater. The temperature signals of the thermocouples were recorded by a Bristol multi-channel recorder. Good results were recorded on the first test run. With both vapor transfer valves open, a steady state temperature difference of $3^{\circ}C$ between the surfaces of the two wicks was recorded at a power input of 69 watts (equivalent to 8500 watt/m^2 of wick area).

This result could not be repeated the next day. Performance of the device deteriorated during the following days. Measurement errors, thermocouple separation and other causes were suspected and checks performed to verify the integrity of the experimental arrangement and equipment. Every possible precaution was taken to eliminate suspected causes for the nonrepeatability of the results. The device was several times disassembled for internal inspection. Wick performance, thermocouple location, thermocouple electrical integrity and freedom of the vapor flow passages from obstructions were checked. Masspectrometer-helium leak checks, after reassembly, where performed. Evacuation was combined with heat addition (bake out) to free the cavities of absorbed noncondensible gases. Vacuum and/or heat was applied to the water, used for charging of the simulator for extended periods of time in order to make sure that dissolved air would be expelled from the water. Yet, the results of these tests were erratic and generally unsatisfactory. The observations made led more and more to the conclusion that, in spite of all precautions taken, noncondensible gases must be present within the cavities of the device. A further modification was then made in order to verify this theory. A second tube, as shown (in phantom lines) in Figure 20 was inserted into the lower cavity of the device. The test set-up was modified as shown in Figures 22, 23, and 24.

Figure 22 shows a view of the device placed in the vacuum chamber prior to insulation. Figure 23 shows an overall view of the experimental arrangement, including instrumentation, recorder and other accessories. As Figure 24 shows, valves were inserted into the fill and the newly added evacuation tube. A connection from the fill tube to the titration tube and a storage bottle with distilled water was maintained during the experiment. Both, the top of the titration tube and the distilled water bottle were connected via a cold trap to a vacuum pump for degassing of the water. The evacuation tube could be alternately connected to the rough vacuum pump via the cold trap or to a diffusion pump, which could provide a hard vacuum. Other accesories shown in Figure 24 include a source of nitrogen for pressurization and connections to the vacuum pump for evacuation of the bellows chambers which operated the vapor transfer valves. An ice-water bath with a submerged pump supplied chilled water to the copper block heat sink.

This arrangement permitted venting and addition of working fluid during the operation of the unit. It was initially assumed that venting during operation would cause significant losses of water vapor and that working fluid would have to be added every time the unit was vented. It was found later that this water addition was not necessary every time venting took place.

Initial experimentation was performed at room atmosphere, with the bell jar raised and without thermal insulation. For experimental purposes, the water fill line was left disconnected and the water fill valve adjusted to permit admission of a trace amount of room air into the device. After an overnight shut down, temperature differences between thermocouples No. 3 and 4 of 25°C were recorded when power and chilled water were applied. Momentary opening of the vent valve for about 2 seconds resulted in an immediate response. The temperatures recorded by thermocouples 3 and 4 approached each other to within less than 1°C. This good performance then deteriorated over a period of approximately 5 to 6 minutes to a temperature difference of about three (3) degrees at which time

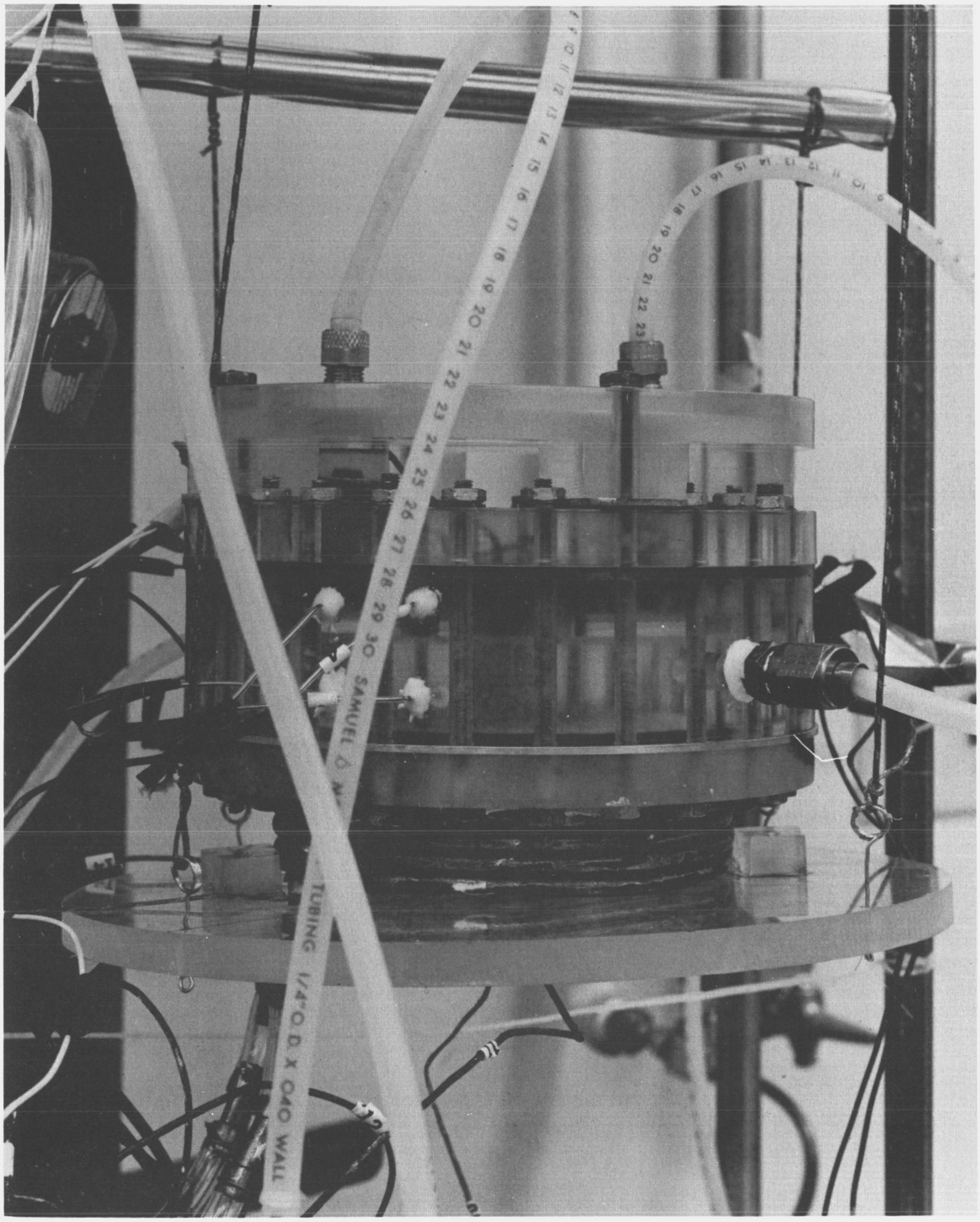


Figure 22 CLOSE-UP VIEW OF VARIABLE THERMAL CONDUCTANCE SPACE SUIT SHELL SIMULATOR

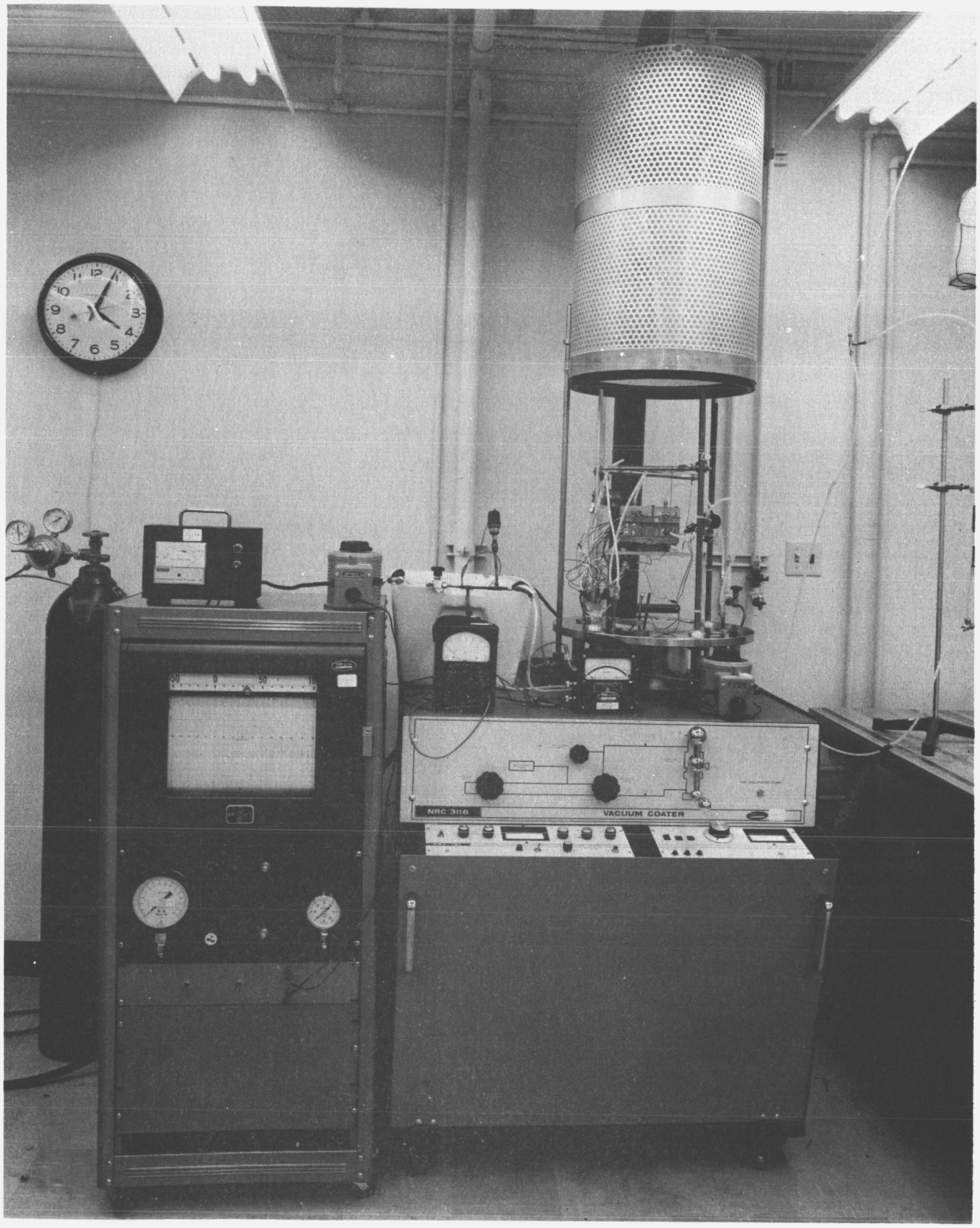


Figure 23 VIEW OF TEST ARRANGEMENT FOR VARIABLE THERMAL CONDUCTANCE SUIT SHELL SIMULATOR

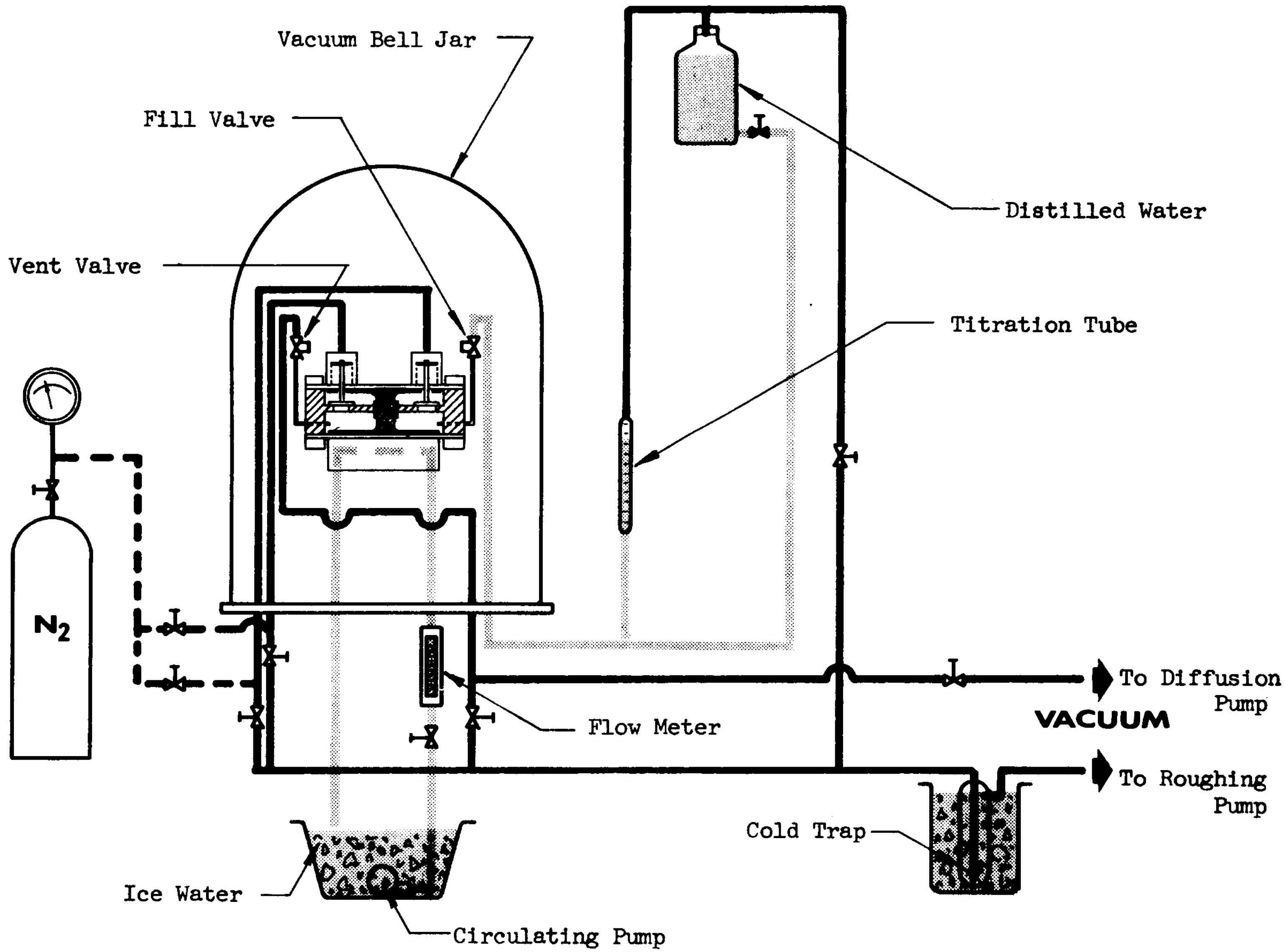


FIG. 24 DIAGRAM OF TEST ARRANGEMENT FOR VARIABLE THERMAL CONDUCTANCE SUIT SHELL SIMULATOR

momentary opening and closing of the vent valve immediately restored the less than 1°C temperature difference.

Tests performed with a small amount of noncondensible gases in the device also demonstrated that it would still operate as a heat pipe, but one of very low efficiency. With enough noncondensible gas in the cavity to result in a temperature difference between thermocouples No. 3 and 4 of 15°C, temperature difference was insensitive to heat input level and essentially constant when heat input was increased from 15 to 45 watts.

This experimentation established that noncondensible gases within the cavities of the simulation device had been responsible for the previously experienced difficulties. It was also observed that either very little water vapor was lost in the process of venting or that the device was relatively insensitive to the amount of working fluid.

The test arrangement was then completed for performance testing. The vent valve, which had been originally manually operated, was replaced by a solenoid valve which could be operated from the outside of the bell jar. Insulation was applied, consisting of 20 layers of crinkled aluminum foil. A thermocouple was placed between the outermost and the second layer of the insulation, facing the electric resistance heater. Readings from this thermocouple were used to verify that the insulation provided adequate protection against heat losses and that a nearly adiabatic condition was maintained. Data taking tests were then performed with the bell jar evacuated to 10^{-6} torr.

During these tests, data were recorded for the temperature of the liquid vapor interfaces on the wicks and the heat input and heat rejection plate, as represented by the temperature readings of thermocouples 3 and 4, and 2 and 5 respectively. Several levels of power input were used. The vapor valves were opened and closed and the response time vs. temperature readings recorded.

The results of these tests are summarized and presented in Figures 25 and 26. Figure 25 shows a near independence of the temperature difference between the surfaces of the wicks (thermocouples No. 3 and 4) from heat flow. The increasing temperature difference with increased power input between the external surfaces (thermocouples No. 2 and 5) is obviously due to the conductive resistance from the outside surfaces to the liquid-vapor interface on the wicks. This is indicated by the increase of the temperature difference between thermocouples No. 2 and 3 and 4 and 5. As the resistance to heat conduction from the outside surfaces to the surfaces of the

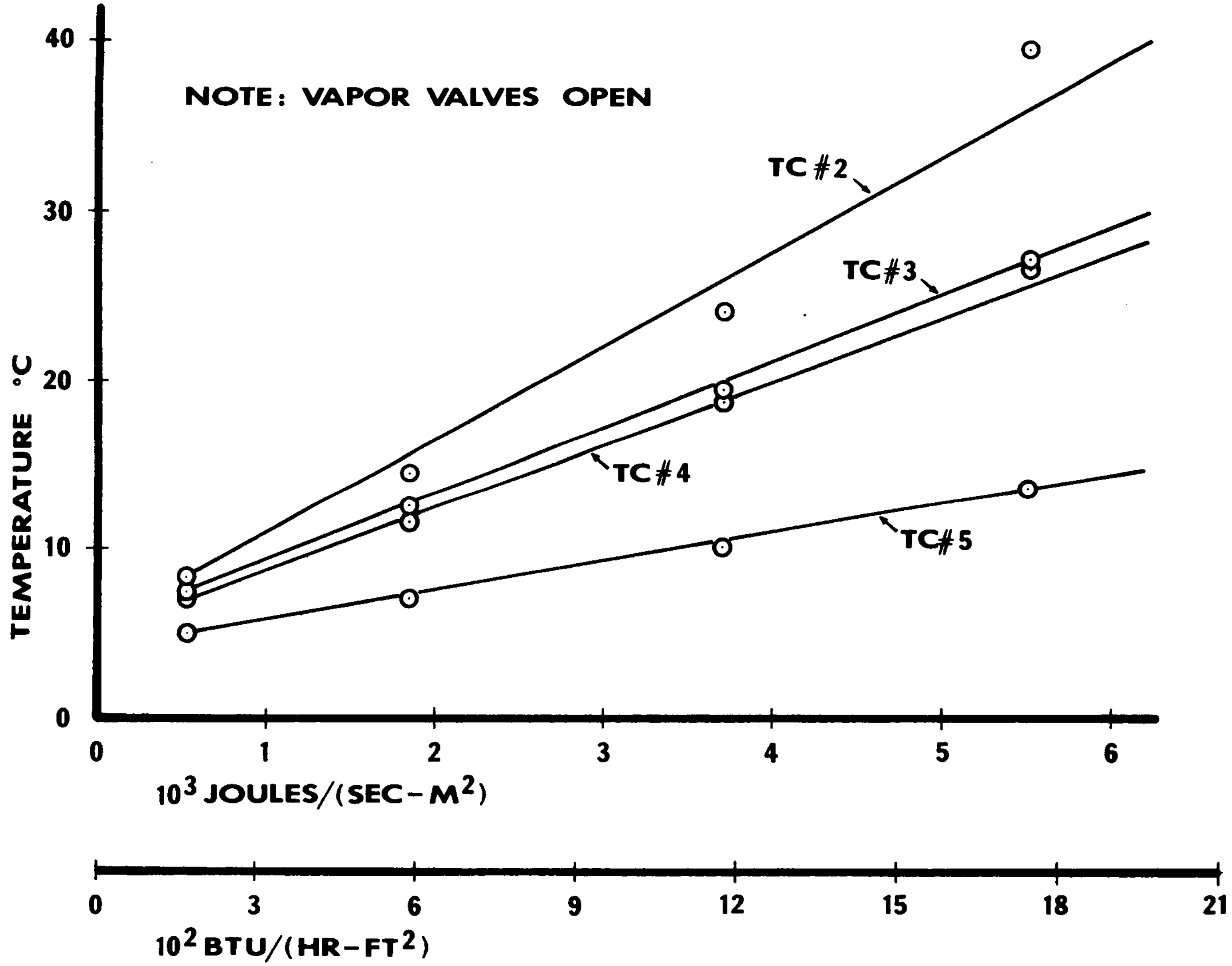


FIG. 25 TEMPERATURE DISTRIBUTION VERSUS HEAT FLOW ON VARIABLE THERMAL CONDUCTANCE SUIT PANEL SIMULATOR

POWER INPUT RATE: 600 JOULES/(SEC – M²)

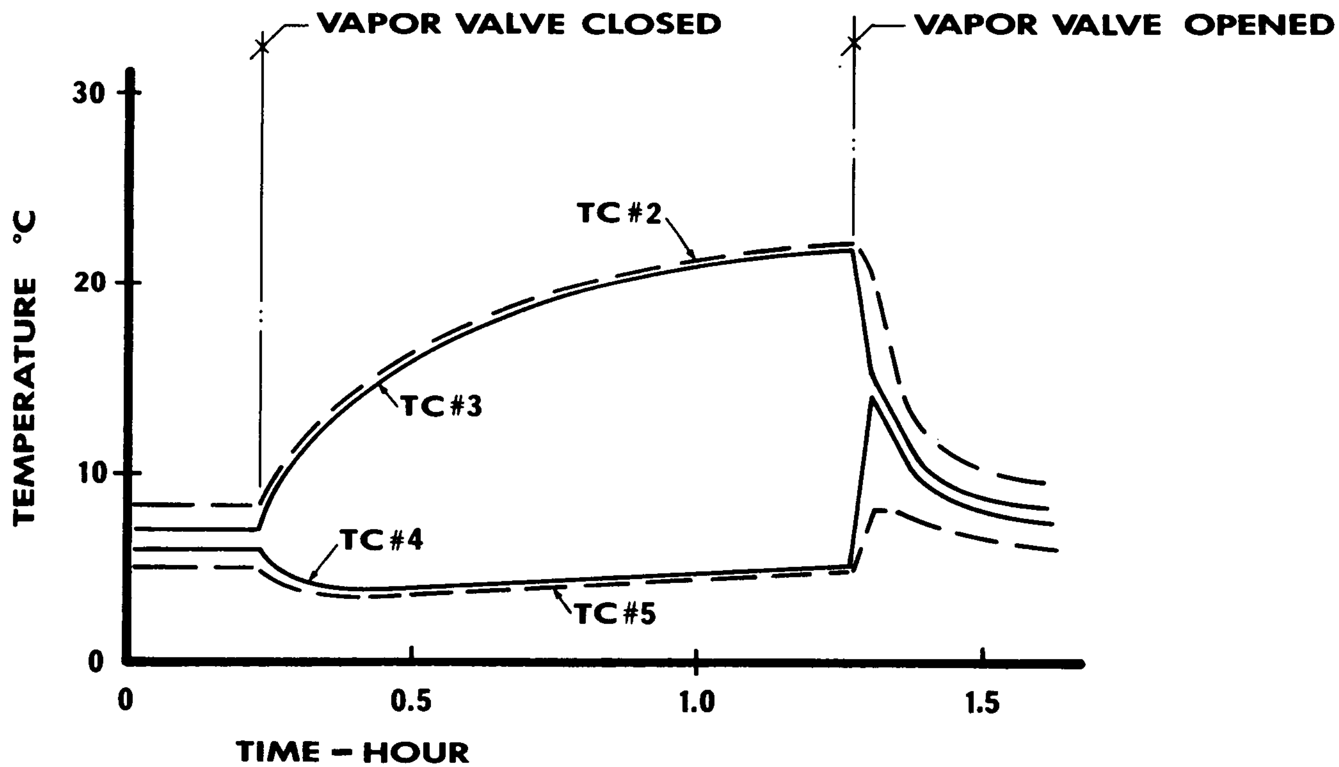


FIG. 26 RESPONSE TO CLOSING AND OPENING OF VAPOR FLOW VALVES ON VARIABLE THERMAL CONDUCTANCE SUIT PANEL SIMULATOR

wicks is constant, the temperature difference must be proportional to heat flow.

The response to the opening and closing of the valves shown in Figure 26 is almost immediate. No difference could be observed between operation or response with one or two vapor valves. The increase in temperature difference with time with closed vapor valves is due to storing of heat within the device, because of the nearly adiabatic condition of the system. With the valves closed, heat transfer across the device from the resistance heater to the heat sink block is so low, that a continuous temperature rise on the heater side must occur. It should be noted that with the vapor valves closed, the temperature on the heat input side increases drastically, while the cold side temperature hardly changes. That the heat flow across the device is drastically reduced is also apparent from the very small temperature differences between thermocouples No. 2 and 3 and thermocouples No. 4 and 5. As these thermocouple locations are separated by a constant thermal resistance and temperature difference is proportional to heat flow, the small temperature difference demonstrates that only very small heat flow takes place.

The results of this series of tests confirm that the concept of a variable conductance heat pipe is correct and sound. It was proven that the presence of noncondensible gases in this or any other heat pipe is a critical detriment to proper heat pipe function. It was demonstrated that noncondensible gases were present in the device and the cause of unsuccessful operation.

The source and the nature of these noncondensible gases has not been established at this time. The most likely potential sources are the construction materials of the simulating device itself. The extreme care which had been taken to prevent leakage into the device and to outgas the working fluid makes presence of air in the device a less likely candidate. A preliminary test on outgassing of Plexiglass seems to confirm this assumption. This test is described in the "Materials Research Report-First Year" TRW Systems, No. 06462-6003-R000, September, 1967, prepared under the same contract.

DESIGN AND DEVELOPMENT OF THE VARIABLE CONDUCTANCE SPACE SUIT PANEL PROTOTYPE

A prototype panel assembly essentially conforming with the schematic cross section of the variable conductance space suit shell shown in Figure 7 was designed and fabricated. In order to be able to fabricate this panel assembly it was necessary to develop techniques for the fabrication of a fiberglass pressure shell with integral heat pipes and a fiberglass two-chamber heat pipe. Problems of attaching wicks to the interior surfaces of the pressure shell had to be resolved. A concept for the design of a two-chamber heat pipe, incorporating thermal insulation had to be generated.

Development of the Heat Pipe Pressure Shell

As previously discussed in this report, the pressure shell was to be modified such as to provide high thermal conductance by lining all interior cavity surfaces with wicks, which when soaked with water would make these cavities function as heat pipes.

A technique for fabrication of a pressure shell with all internal surfaces lined with Refrasil wick material is described in detail in the "Materials Research Report-First Year", TRW Systems No. 06462-6003-R000, September 1967, prepared under the same contract. Summarizing the fabrication technique, it shall only be stated here that it was impractical to insert and attach wicks into the small and long tubular channels after fabrication of the fiberglass panels. Instead, the tubular channels are built up around removable teflon mandrels beginning with the wick, followed by layers of polyester-coated mylar and fiberglass. The wick-lined tubes so fabricated are then assembled between two layers of fiberglass sheets and the whole assembly bonded by heat and pressure.

Figure 27 shows in the center a sample of the fiberglass pressure shell originally proposed by the NASA/Ames Research Center. On the left side is a view of the modified pressure shell with wicks included into trapezoidal tubular cavities.

In order to verify the effect of the use of the internal cavities as heat pipes, a sample section of this panel was closed off on the edges and a tube for filling and evacuation bonded to it. The heat pipe panel thereby formed was evacuated and a measured amount of water added to saturate the wicks. A heater was attached to one of the

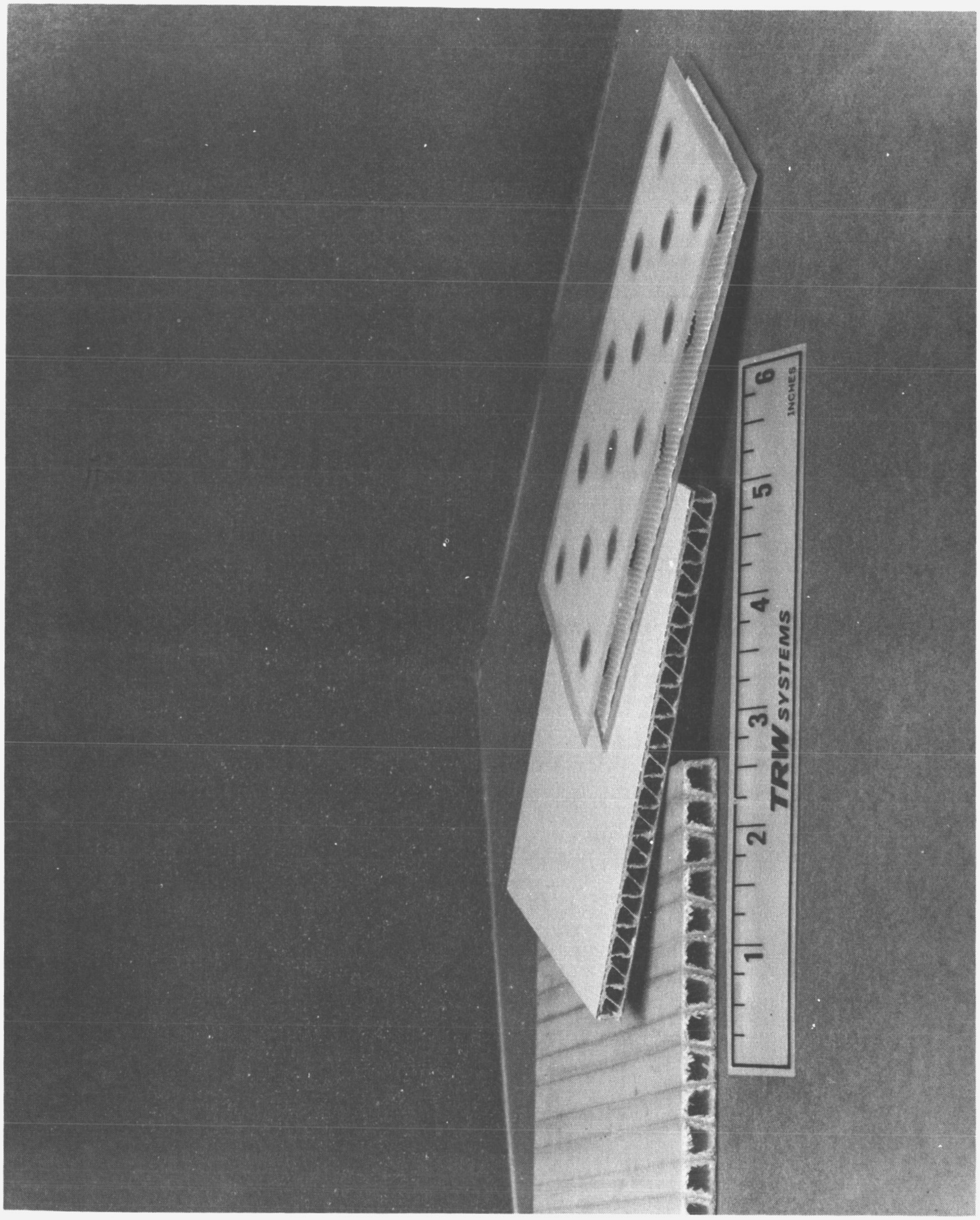


Figure 27 SAMPLES OF FIBERGLASS SUIT SHELL PANELS

external surfaces. The other external surface was brought into contact with a water-cooled copper block. Silicon grease Dow Corning #340, was used for improved thermal contact. Thermocouples were attached to the external surfaces. The tests were performed in a vacuum bell jar at 10^{-6} torr and 20 layers of crinkled aluminum foil for insulation. Various levels of power were applied to the electric heater and the resulting temperatures recorded.

The panel was then removed, the evacuation tube opened up and the water used as working fluid permitted to evaporate. The test was then repeated with the fill tube open to the vacuum of the bell jar. The same power inputs were used and the temperatures on the two surfaces recorded.

The data so obtained provided a comparison of the temperature gradients obtained with the panel working as a heat pipe or with the thermal conductance of the panel structure providing heat transfer. The results of these tests are plotted on Figure 28. Significant improvement of heat transfer resulted from the heat pipe action, although the effects are less drastic than those obtained with the pressure panel simulator heat pipe shown on Figure 17. Obviously, the fiberglass construction results in higher conductance resistance on the heat input and heat removal side of the panel, when compared with the aluminum plates used on the simulator. This resistance is constant and the effect of change in resistance across the internal cavities is therefore less pronounced.

Development of the Variable Conductance Space Suit Shell

The functional feasibility of the variable conductance space suit shell was demonstrated with the help of the simulation device, described in the preceding chapter. This simulation device demonstrated that heat flow could be stopped or promoted by closing or opening of vapor valves in a two-chamber heat pipe.

The concept of the variable conductance space suit shell is that of a two-chamber heat pipe, with both chambers separated by a thermal insulating layer. Heat transport between the two chambers would be by heat pipe function through a limited number of tubular connections penetrating through the insulation layer and providing vapor flow passages and wick passages between the two chambers.

The geometry of a space suit requires that the two chambers be in the shape of flat, large, hollow panels. These flat and hollow panels must be designed to permit evacuation, as necessary for removal

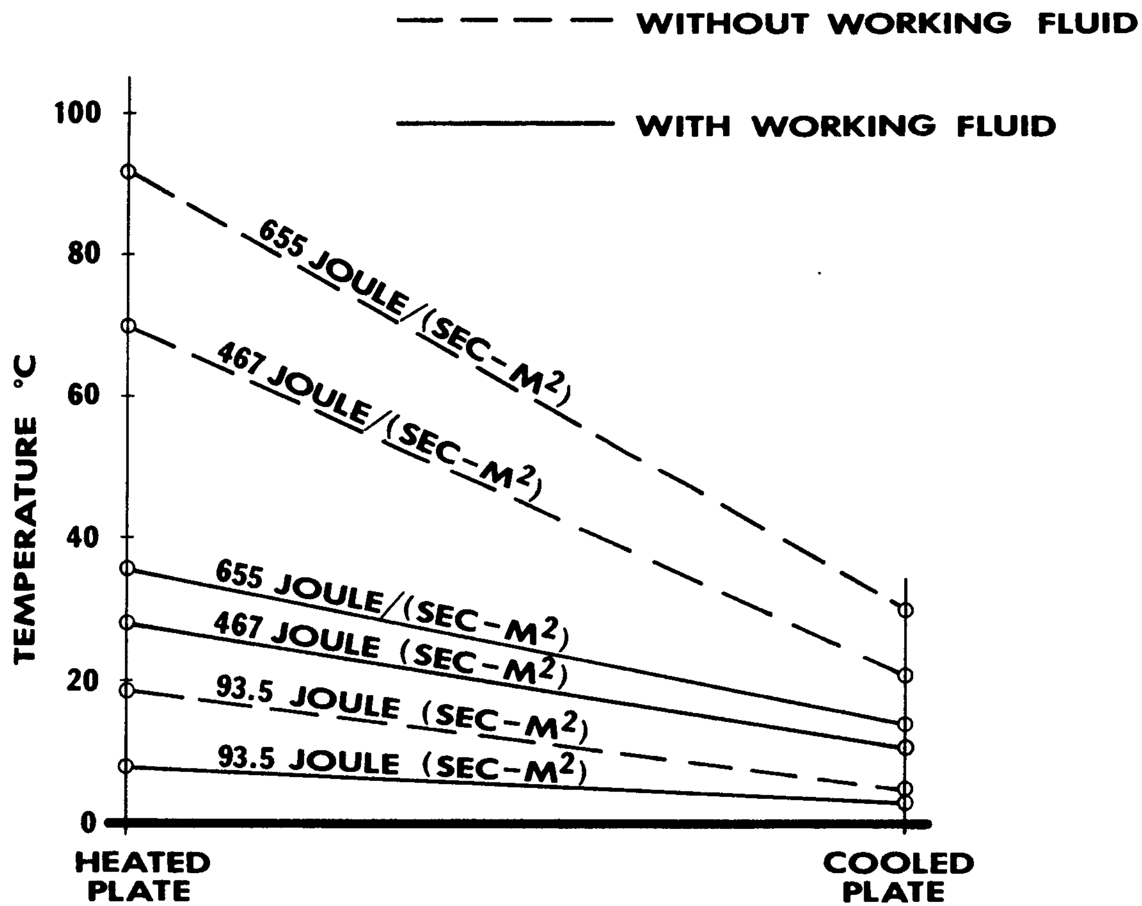


FIG. 28 TEMPERATURE GRADIENT ACROSS SUIT PANEL PRESSURE SHELL

of noncondensible gases, without collapsing. The operating pressure within these chambers will be very low, for example, with methyl alcohol as a working fluid, only 29.6 mm at $0^{o}C$. While these panels during extravehicular activities would not be exposed to external pressures, they are required to withstand sea level and space craft cabin pressure during storage and check out.

In the outer suit surface heat diffusion parallel to the suit surface by heat pipe function is required for temperature equalization between irradiated and non-irradiated sections of the space suit shell. This precludes the use of the corrugated reenforcements used for the pressure shell of the space suit which result in parallel tubular cavities, providing lateral heat flow predominantly in the direction of the tubes.

A concept was generated to construct the panels for the two chambers of the variable conductance space suit shell of two parallel sheets separated by equally spaced supporting columns. (See panel on right of Figure 27). These columns would reduce the free span between support points. Several experimental samples of such panels were fabricated. After a reasonable learning period in assembly of panels and lining their inside surfaces with wicks, a sample panel with a triangular arrangement of columns spaced 30 mm on centers was selected for vacuum testing. It was equipped with Refrasil wicks bonded to the internal surfaces. A section of 150 mm by 90 mm was sealed on all edges with fiberglass strips and a copper tube for evacuation bonded to it. The panel was then connected to a diffusion pump and evacuated to 10^{-6} torr. While slight inward bulging between the columns could be observed, it withstood normal atmosphere pressure without collapsing, leaking, or wick separation. This concept was, therefore, selected for prototype fabrication of a suit panel.

Two panels of this type, lined with wicks on all internal surfaces, separated by thermal insulation and interconnected with wick passages and vapor flow passages will represent the variable thermal conductance heat pipe for the space suit.

The details of fabrication and assembly of the various suit shell panels are described in the "Materials Research Report-First Year", TRW Systems No. 06462-6003-R000, September, 1967.

Figure 29 shows a photograph of a complete variable thermal conductance space suit shell prototype panel which has been fabricated for purposes of thermodynamic testing. This prototype panel conforms

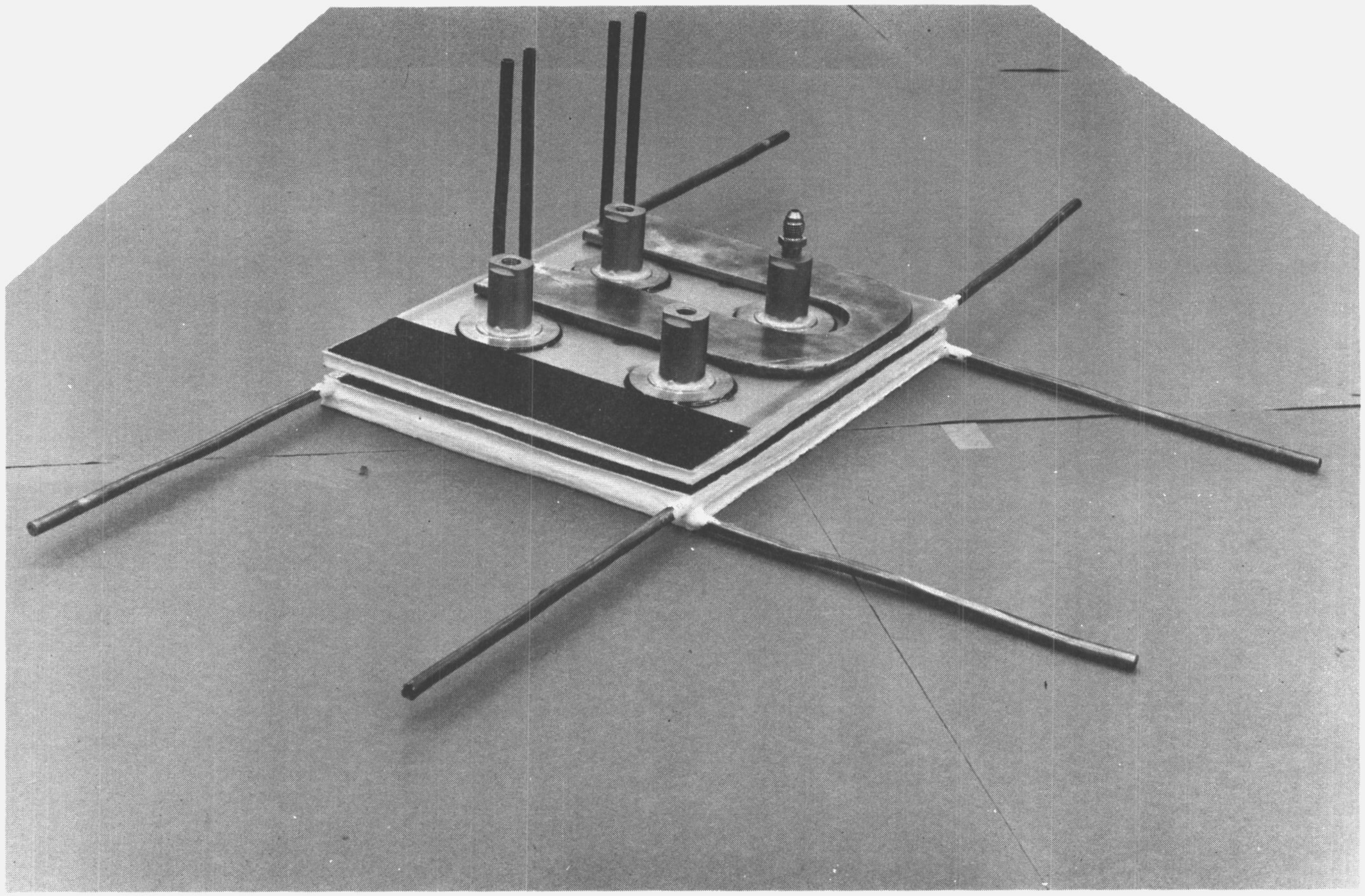


Figure 29 PROTOTYPE PANEL OF VARIABLE THERMAL CONDUCTANCE SPACE SUIT SHELL

in its internal arrangement essentially to the cross section shown in Figure 7. The only exception is that the bellows for operation of the control valves are located externally.

Some devices required for the performance of the experiment have been added to the panel. The horseshoe shaped device on top of the panel is a heat exchanger through which chilled water will be circulated during tests for simulation of heat rejection from the external suit surface by radiation to space. The black rectangular area is an electrical resistance heater, bonded to the surface for simulation of external radiative heat inputs. Another resistance heater on the underside of the panel and not visible in the picture will simulate metabolic heat input from the human skin. The dark circumferential line is the gap for thermal insulation between the two chambers of the heat pipe. The horizontal tubes are fill and vent tubes. Those not necessary will be cut and sealed. The four vertical tubes are for inlet and outlet of chilled water to the heat exchanger used as heat sink.

The prototype is designed for evaluation of heat flow rate with valves open or closed as well as lateral temperature equalization in the external chamber of the two chamber heat pipe.

RECOMMENDATIONS

The activities described in this report have demonstrated functional feasibility and practicability of fabrication of the variable thermal conductance space suit shell. Promising possibilities exist, therefore, for the development of extravehicular space suits which take advantage of radiation from the external suit surface as the major method of metabolic heat rejection. Further activities are recommended, which include the following major areas of research:

- The sources of noncondensible gases which have caused so much difficulty in performance of experimentation shall be identified and eliminated.
- Techniques for transfer of body heat from the human skin into the variable conductance space suit shell shall be investigated.
- The possibilities for improvement of conductive heat transfer from the outer surfaces of the space suit shell heat pipes to the liquid-vapor interfaces at the wick surfaces shall be investigated.

- The selection of working fluids shall be reviewed. Preferably water should be used as the only working fluid in a space suit. This will require development of freeze preventing techniques.

- An improved concept of the thermal control space suit shell shall be developed. This shell shall be of less complexity and shall incorporate passive humidity control and possibly means of auxiliary heat rejection.

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